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COGENERATION FEASIBILITY STUDY

Comparison of Rankine vs Stirling Engines Adapted to a Solid Fuel Combustor

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COGENERATION FEASIBILITY STUDY
Comparison of Rankine vs Stirling Engines Adapted to a Solid Fuel Combustor

Prepared by

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November 30, 1983

Prepared for

Montana Department of Natural Resources and Conservation
1520 East 6th Avenue, Helena, Montana 59620
Biomass Utilization and Cogeneration Program
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1. Barber-Nichols Feasibility Study
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3. S.P.S. Power Without Fuel
4. OMAT Technical Bulletin No. 252W

B. Stirling Engine

1. Stirling History - W.H. Percival, United Stirling, Inc.
2. Martini Engineering
3. Design of a Low Pressure Air Engine for Third World Use
4. Biomass Fueled Stirling Engine
5. 1 KWe Free Piston Stirling Engine
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C. Sodium Heat Engine

1. Research on the Sodium Heat Engine
2. Solar Residential Total Energy System Using the Sodium Heat Engine

D. The Scroll Machine

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B. United Stirling

C. Barber-Nichols Engineering

D. Power Generation Through Waste Heat Recovery

1.0 SUMMARY

This preliminary study was done to compare the technical and economic feasibility of adapting the Stirling engine or Rankine cycle engine to the Boucher Combustor for cogeneration. Boucher's combustor is a high efficiency, low emissions solid fuel system utilizing a newly invented combustion process. This cogeneration system would use the engines as a topping cycle to produce electricity and use the waste heat from the engine for space heating and domestic hot water, to be applied as a self-contained energy system for homes and commercial buildings. My investigations indicate that it is technically feasible and economically justifiable to develop such a cogeneration system and that there are several other types of engines that may soon prove workable. At present, the Rankine cycle engine appears most practical, although the free-piston Stirling engine is under development and may be more feasible than the Rankine engine in the future. I have decided to continue my research before deciding which technology would be best suited for my combustor.

2.0 INTRODUCTION

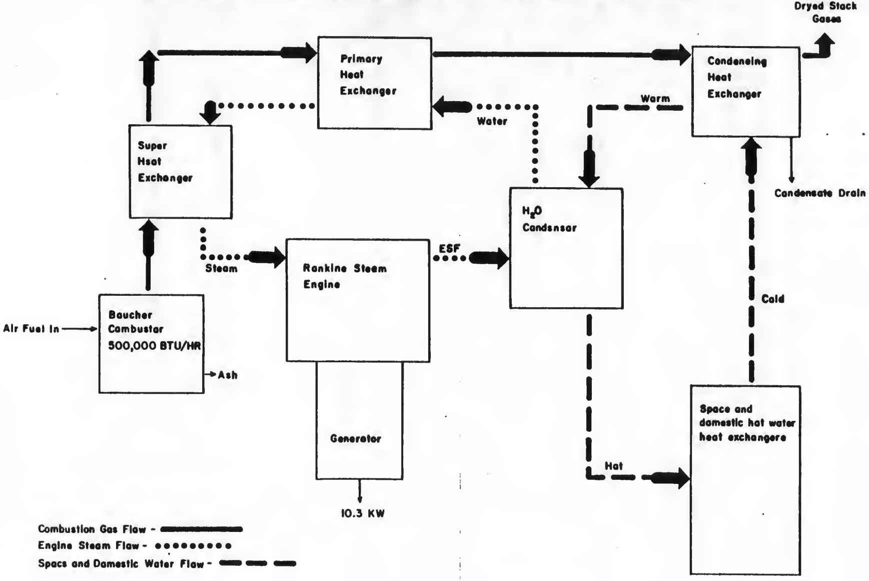
A successful self-contained cogeneration system using biomass fuels requires a clean and efficient combustor to convert the stored energy into useful thermal energy. I currently am working on a prototype 500,000 btu/hour combustor that will burn wood chips or biomass pellets to produce hot water for space heat and domestic use at a motel in Missoula, Montana. The prototype will be completed and tested in May 1984. I anticipate the emissions of particulates, hydrocarbons and CO will be one half or less of the Missoula County emissions standard for residential wood burning devices. I expect to achieve 98% combustion efficiency, 92% thermal transfer efficiency for an overall efficiency of 90%. The basic concept is to get the combustion process completed prior to extracting the energy, than extract the energy from the combustion gases. The combustor has a condensing heat exchanger that cools the combustion gases to 100°F, causing 95% of the water in the gases to condense out. This gives three advantages: It increases efficiency approximately 18%, acts as a natural flue gas scrubber and eliminates flue fire danger. Condensing this water will recover 2,162,635 btu's/day, about 26% of the motel energy requirement. One hundred ninety one gallons of condensate will be produced per day. I have invented a burner head that basically gasifies the fuel, separates the gases from the char and then combusts the gases and the char separately. This head can be modified for several combustion characteristics. This combustor would be the prototype to the one used in the cogeneration system and expected to be perfected prior to the final design of the cogeneration system. My research indicates the same combustor modified can burn coal and eliminate sulfer dioxide emissions.

I performed a literature search on the Stirling engine and the Rankine engine to determine the research progress and the current uses of both engines. The Stirling engine has more research programs in progress but virtually no practical commercial application at this time. It appears most of the funding for the Stirling research comes from government programs, with the main thrust toward automotive and solar applications. There is also a limited amount of research in electrical generation using liquid and solid fuels. Although research continues on the Rankine cycle engines, there are commercial systems on the market using this technology. The research is mainly in the solar and geothermal application. The commercial application for the organic Rankine cycle engine is primiarily bottoming cycle cogeneration using industrial waste heat. And of course, the steam Rankine cycle engine has been in use many years. During this study I found information on the Sodium Heat engine (SHE), the Scroll Machine and thermovolt generator all of which could have good potential in a topping cycle cogeneration system. At present the steam Rankine cycle engine is best suited for my cogeneration system because it is the most economical to develop and manufacture, and the most technically perfected. Consequently, a production prototype can be designed and built faster and more economically than the Stirling engine. My study indicates that new technologies are rapidly advancing and these could prove to be superior to the Rankine cycle engine in adapting to my combustor. Given the state of the current research, I decided to make a more thorough investigation into these other engines and monitor the success of the existing research for six to twelve months before making a final determination which engine would be best for my cogeneration system. Much of the research is directly applicable to cogeneration and could save substantial research money for my system.

2.1 RANKINE VS STIRLING

	Rankine	Stirling
Developmental cost	$270,000	$500,000 plus
Life	200,000 hrs	10,000 hrs
Maintenance	Periodic inspection Minimal maintenance	Working fluid leakage Many moving parts which contributes to failures
Cycle Efficiency	13%	15% - 25%
Commercial Status	Many years of use and commercialization. Readily adaptable, can use wide range of combustion gas temperatures.	Successful prototypes but no commercial use. Marginally adaptable, combustor may require modification to meet engine requirements. Can use only narrow temperature range of combustion gases.
Development Stage	Technology advanced to where production prototype can be designed.	Research required.
KWe Sizing	10 KWe plus	1 KWe to 15 KWe
Quality of Electricity	Very good	Unknown
Successful prototype	Yes	Questionable
Experienced Manufacturers	Several	None
Production Cost	$10,000	Unknown

Figure 2.2
Block Diagram Of The Boucher Combustor And Rankine Engine Cogeneration System



3.0 COGENERATION PHILOSOPHY

The concept of a self-contained cogeneration energy system using solid fuels, solar energy and liquid fuels for residential and small commercial buildings has existed for some time. The technology and the market exists to develop such a system. If such a system were developed and used, it would take part of the utilities out of government control and place it in the private sector. The energy system would be bought like an automobile; financed, maintained and serviced by the consumer. The manufacturer would give warranties; provide service centers and develop new and improved models. With the several potential engines the consumer would have a choice of a "Ford," "Chevy" or "Toyota." The combustor would be efficient and nonpolluting, so there would be no environmental impact studies, protest over new central facilities or court battles over power lines. The central power facilities would be augmented by these cogeneration systems. It would give the consumer a sense of independence from the utilities.

4.0 ENGINES

This study led to my discovery of several engines other than the Rankine and Stirling engines that may be feasible to adapt to my combustor as a cogeneration system. They are the Sodium Heat engine - Ford Motor Company, Scroll Machine - Arthur D. Little, Inc., Thermoelectric - Photic Corporation.

4.1 RANKINE CYCLE ENGINE

After I decided the Rankine cycle engine currently is more readily adaptable to my solid fuel combustor, I contracted with Barber-Nichols Engineering, Arvada, Colorado to provide a preliminary design study and make cost estimates. This study is included in this report. The study shows that adapting a Rankine cycle engine to my solid fuel combustor is technically feasible with no major problems. It also would be considerably less costly to develop than the Stirling engine. Barber-Nichols would be interested in manufacturing the engine and generating system. There are several other companies with the capability to manufacture production models. Barber-Nichols indicates it could design the system,build and test a prototype for $270,000, not including the cost of the combustor. That is considerably less than the cost in developing the Stirling engine. The operating life of the engine is estimated to be at least 200,000 hours(22 years) with minimal maintenance. The high speed (60,000 rpm) of this engine would produce high quality electricity. There are no current projects in progress that parallel my proposed cogeneration system, so there would be no new design data that could be taken advantage of.

4.2. STIRLING ENGINE

I rejected the Stirling engine for three primary reasons. The Stirling engine is not easily adaptable to my combustor because of its high temperature requirements and it cannot use the lower combustion gas temperatures as the Rankine engine can. The operating life is low compared to the Rankine engine, no more than 10,000 hours. There are no commercial manufacturers.

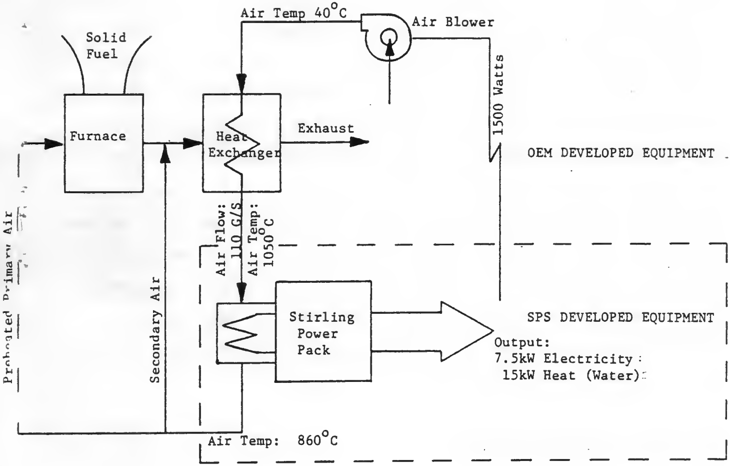
There are two types of Stirling engines, the kinematic and the free-piston. The kinematic is more developed and is the primary engine being worked with to date, and appears to be close to having a practical commercial use. The free-piston engine is a high-tech engine and commercialization is not in the near future, although it appears to have good potential. The free-piston engine may have more potential in a cogeneration system than the kinematic because of its simplicity and lower maintenance requirements. There are several Stirling engine research projects in progress that parallel my proposed system and the design data could substantially reduce my developmental cost. United Stirling, M.T.I. and Sun Power are working on biomass fueled Stirling engines. I could not determine the current degree of success of these programs.

4.2.1 V160 STIRLING ENGINE

The V160 Stirling engine by Stirling Power Systems, Ann Arbor, Michigan, appears to be the closest unit to commercialization that is available that could be adapted to my combustor. I went to Ann Arbor to look at the engine and talk with the engineers, and concluded that it is not practical to adapt this engine to my combustor. The V160 requires 187 cfm of air at 2000°F to provide 7.5 KW of electricity and 15 KW hot water, and ejects this air at 1500°F. Air is the heat transfer medium. To make the system efficient, I would need to be able to

V160 Stirling Engine

By Stirling Power Systems



use the exit air from the engine for combustion air for the combustor or extract the heat with an additional heat exchanger. If the exit air is used for combustion, a 100,000 btu/hour combustor would provide enough energy to drive this engine, $\frac{(22.5 \text{ KW} \times 3400)}{80\% \text{ eff}} = 95{,}625$ btu. A 100,000 btu/hour combustor requires only 20 cfm of air; 600°F is the optimum air preheat temperature; the combustor's efficiency drops substantially as excess air increases. Therefore, it is not efficient to use this engine's exit air for the combustion air. To provide another heat exchanger for this hot air would complicate the overall system and be expensive. If there were a heat transfer medium cycling in a closed system through the combustor heat exchanger and V160 it might solve this problem, but it would mean redesigning the V160. The temperatures are too high for economical alloys. U.L. standards rate 304 s/s at 1200°F and 316 s/s at 1600°F. Ceramic tubes might be used. The life of this engine is 5,000 hours which is far too short for continuous cogeneration (1 year = 8,760 hours). At this time there isn't a manufacturer for the engine. The V160 is projected to be marketed in 1985 at a cost of $2600 per engine. Currently prototypes are available at a cost of $27,150.

4.3 SODIUM HEAT ENGINE (SHE)

The SHE was invented at Ford Motor Company, a result of research on a sodium-sulfur battery. The SHE is an electric generator that converts heat energy directly into electricity. Experimental engines are about 20% efficient but 30-40 percent efficiencies are expected in the future. It is particularly suited for a cogeneration system using any heat source. SHE has many very good advantages; no moving parts and only a single working fluid, no expensive materials, high efficiency, no variation in efficiency with size, high power density and light weight. I have included two papers on the SHE in my report.

4.4 THE SCROLL MACHINE

The Scroll machine is based on an old concept. It can be used as a pump, compressor or an engine. The Scroll machine can be used as a Rankine Cycle or the Brayton Cycle. Arthur D. Little, Inc. is developing the Scroll Machine and has shown an interest in looking at applying the Scroll Machine to a solid fuel combustor. I have included in this report an article from Mechanical Engineering.

4.5 THERMOVOLTAICS

Photic Corporation has developed and is marketing a thermal generator that uses a solid fuel combustor for its thermal energy. Thermovolt^{TM} is a low electric output generator designed for remote areas and could have applications in residential homes.

5.0 SAFETY

I have not found any safety problems with the Rankine engine or the Stirling engine. The Rankine engine would operate at relatively high pressure and would be required to meet ASME safety standards, which poses no problems. According to the Montana state boiler inspector, the system would be classified as a non-fired pressure vessel because it takes its energy from the combustion gases and not directly out of the combustion chamber. Therefore no licensed operator would be required to operate the system. I could not find any safety problems with the Stirling engine. Apparently some safety tests have been conducted but I did not find the documentation. The combustor would have to meet U.L. safety standards, which it could do.

6.0 SYSTEM COSTS

It was very hard to determine production costs for the engines. A rough guess to design, build and test a free-piston prototype with my combustor would be $500,000. If current research projects are successful, the cost could possibly be reduced to $100,000 by 1985. At present, I cannot assume the first prototype would be successful. The preliminary cost estimate for the Sodium Heat engine is $1,000 per KW for a "few units per year" and $250 per KW for production. I could not determine developmental cost to adapt the SHE to my combustor. I have no cost estimates whatsoever for the Scroll Machine.

Barber-Nichols cost estimates for the Rankine engine is $270,000 to design, build and test a prototype. Production of a thousand units per year, the cost would be $20,00 per unit and possibly get down to $10,000 per engine. The test prototype combustor would be $15,000, the production unit would be $6,000 to $7,000. Therefore an estimated cost of $16,000 to $25,000 for a production model producing 10.3 KWe and 313,500 btu/hr of hot water or approximately 92 KW_{th}.

The 500,000 btu combustor would burn 9.6 tons of wood chips a week. At $25 a ton for chips at 50% moisture, the fuel cost is $239 a week. For the Reserve Street Inn in Missoula, Montana that would be $269 a week savings. With 95% of the moisture condensed out of the combustion gases the fuel savings will be $317 per week.

6.1 THERMAL ENERGY VALUE

Daily energy consumption of the Reserve Street Inn, Missoula Montana, December 6,1983 to Jan 6, 1984.

1st Building		1023.2 KWh	1.53 MCF
2nd Building		674.8 KWh	.98 MCF
	Total	1698.0 KWh	2.51MCF

1698 KWh x 3413 btu/KWh = 5,795,274

2.51 MCF x 929,980 btu/MCF = 2,334,249

Total = 8,129,523 btu's/day

Wood chips having 50% moisture, H_2O wt/wood wt plus H_2O wt

8,750 btu/lb dry x .50 = 4,375 btu/lb

$$\frac{8{,}129{,}523 \text{ btu/day}}{24 \text{ hr/day}} = 338{,}730 \text{ btu/hr}$$

$$\frac{338{,}730 \text{ btu/hr}}{4{,}375 \text{ btu/lb}} = 77.4 \text{ lbs/hr} = 1{,}858 \text{ lb/day}$$

Combustor is 90% overall efficient.

$$\frac{77.4 \text{ lb/hr}}{.90} = \underline{86 \text{ lb/hr of fuel}}\text{, approximately 1 ton per day.}$$

7 tons fuel x $25/green ton = $168/wk

1698 KWh/day x $.03/KWh = $50.94/day

2.51 MCF/day x $4.6/MCF = $11.55/day

$62.49/day = $437.43/wk

$437.43/wk - $168/wk = $269.43 savings/wk

This assumes wood chips replaces 100% energy used

6.2 ELECTRICAL ENERGY VALUE

The cogeneration system will produce 10.3 KWe with a 500,000 btu/hr combustor.

10.3 KWe x 24 hr/day = 247.2 KWh/day x $.03/KWh = $7.42

$7.42/day x 365 day/yr = $2,708/year

$$\frac{\$10,000/\text{engine}}{\$\ 2,708/\text{year}} = 3.7 \text{ years}$$

$$\frac{500,000 \text{ btu/hr}}{4,375 \text{ btu/lb}} = 114.28 \text{ lbs chips/hr} = 2,742 \text{ lb/day}$$

1.37 ton/day x $25/ton = $34.25/day = $239.75/wk

$239.75/wk with electricity minus

$168/wk without electricity equals

$71.75/wk fuel cost for electricity

$$\frac{\$71.75}{1,730 \text{ Kwh/wk}} = \$.0415/\text{KWh}$$

Considering just the thermal energy cost savings and not the electrical energy value;

$269.43 savings/wk x 52 wk/yr = $14,010.36/yr

Combustor - engine production cost $16,000/unit

$$\frac{\$16,000/\text{unit}}{\$14,010/\text{yr savings}} = 1.14 \text{ years}$$

6.3 CONDENSATE

The 500,000 btu combustor will burn 114.28 pounds of wood chips per hour, with 50% moisture, green basis.

BTUs recovered from condensate

114.28 lbs - 114.28 (.50) = 57.14 lbs/hr wood burned

One pound dry wood with 100% combustion produces .66 lb H_2O.

57.14 lb/hr (.66) plus 57.14 lb H_2O = 94.85 lbs H_2O/hr

94.85 lbs H_2O x 1000 btu/lb H_2O = 94,852.4 btu/hr

95% of the steam condensed

94,852 btu/hr x .95 = 90,110 btu/hr gained from condensate

$$\frac{2{,}162{,}635 \text{ btu/day}}{8{,}129{,}523 \text{ btu/day}} = 26\% \qquad \frac{90{,}110 \text{ btu/hr}}{500{,}000 \text{ btu/hr}} = 18\%$$

(Motel use)

If there appears to be a discrepancy in these calculations it is caused by using the lower heat value of wood, 4,375 btu/lb. The BTUs to vaporize the fuel moisture and the latent heat in the H_2O created by the combustion reaction is recovered as usable energy. Therefore 26% less fuel would be required.

77.4 lbs/hr thermal energy x .74 = 57.28 lb/hr required

4.81 ton/wk x $25/ton = $120.27/wk fuel cost

$437.43/wk - $120.27/wk = $317.15/wk savings

$$\frac{\$16{,}000/\text{unit}}{\$16{,}492/\text{yr savings}} = .97 \text{ yrs}$$

Condensate volume

$$\frac{94.85 \text{ lbs } H_2O/\text{hr}}{8.4 \text{ lb gal}} = 11.29 \text{ gal/hr}$$

11.29 gal/hr x 128 hr/wk = 1,445 gal/wk

75,146 gal/yr

A comprehensive study on this condensate would cost about $30,000.

6.4 DEVELOPMENTAL FUNDS

About $400,000 would be required to get this cogeneration system to production. The amount of funds required to start manufacturing and marketing I do not know yet. Sources of funds are private investment, D.O.E. Biomass Thermochemical Conversion Program, Montana State Renewable Energy Program, Montana Economic Development Program and profits from production of the basic combustor.

I went to Washington,D.C. and talked with the D.O.E. and the E.P.A. (with the aid of Senator Melcher) about my proposed system. Both were very interested. E.P.A.'s interest was the emissions reduction and the condensate disposal. The D.O.E.'s interest was the furnace as a biomass heating system and the cogeneration system. The D.O.E. took a much more in depth look at my system than the E.P.A..

Although both agencies were interested, only generic type research is being funded. My combustor is past that stage. Developmental and prototype funds are not available. The policy is that this funding should come from the private sector.

Although direct funding is not available, D.O.E. expressed a desire to aid me in any way possible and gave me direct access to the program manager of the Energy Conversion Equipment Branch. This department is aiding me in acquiring research papers and also is trying to get my condensate included in an existing research program being conducted by Batelle Laboratory in Columbus, Ohio, studying the oil and gas condensate.

7.0 RECOMMENDATIONS

If the decision had to be made today to select one engine and start designing and building a prototype, the Rankine cycle would have to be the choice, it would have much better predictability in the final results of the system. The odds of having a successful production model after the first prototype are good, but the combustor must be completed and tested and the specifications finalized before the engine can be adapted. This will take about six months. Because of this time lag and because of the rapid development of the free-piston engine and the inadequate study of the SHE and the Scroll Machine they should be evaluated at the time a final decision is to be made. If the funds were available, the three engines should have a preliminary feasibility study done similar to what Barber-Nichols did with the Rankine engine. I would have Ford Motor Company do the SHE, Sun Power, Inc. do the Stirling free-piston engine, and A.D. Little, Inc. do the Scroll Machine. The sizing of the system should be such that it meets the commercial market first. Because they will realize a much faster return in their investment, they would be able to justify the higher cost of the first models. A marketing survey should be done to aid in finding financing. Financial backers should be sought for the prototype, demonstration project and production.

8.0 LIST OF CONTACTS

STIRLING ENGINE CONTACTS

Jet Propulsion Laboratory
Public Information Office
California Institute of Technology
National Aeronautics & Space Administration
Pasadena, California 91109
(213) 354-5011

United Stirling, Inc.
211 The Strand
Alexandria, Virginia 22314
(703) 549-7174

Stirling Power Systems
P.O. Box 1187
Ann Arbor, Michigan 48106
(313) 665-6767

KB United Stirling (Sweden) AB&CO
Box 856
S-201 80 Malmo, Sweden

Energy Research & Generation Inc.
Oakland, California
(415) 658-9785

Department of Energy
Washington, D.C.
Danny C. Lim
Mail Stop GH-068
1000 Independence Ave, SW
Washington, D.C. 20585

Batelle-Northwest
Pacific Northwest Laboratories
Richland, Washington 99352
G.P. Schieffelbein, Manager
Biomass Program Office
(509) 375-2140

Georgia Institute of Technology
Atlanta, Georgia
Dr. Doug Neal
(404) 894-3329

Stirling Thermos Motors
Ann Arbor, Michigan
Dr. Rolse Meijer

Sun Power Inc.
Athen, Ohio
(614)594-2222

Lewis Research Center
Cleveland, Ohio
Frank Kutima, Jr.
(213)577-9416 ext 343

Martini Engineering
2303 Harris
Richland, Washington 99352
(509)375-0155

ORGANIC RANKINE ENGINE CONTACTS

Barber-Nichols
6325 W. 55th Avenue
Arvada, Colorado 80002-2777

Sunstrand Energy Systems
4747 Harrison Avenue
P.O. Box 7002
Rockford, Illinois 61125

Western Consulting Engineers
P.O. Box 784
Mercer Island, Washington 98040

SPS Inc.
Miami, Florida
Vic Worminger
(305)754-7766

Fischer Solar Engine
Phoenix, Arizona
Quinten Adams
(602)264-3919

Ormat
Harvey, Illinois
Anil Prasad
(312)331-4000

9.0 BUDGET

		(DNRC - BPA)	(CONTRACTOR)
1.	Barber-Nichols feasibility study	$ 7,882.58	
2.	Travel		
a.	Air Fare		
	Missoula-Denver-Missoula		$ 416.00
	Msla-Denver-Wash DC-Detroit-Msla		700.00
b.	Meals, lodging		700.00
c.	Secretarial, copying, reports, mail		500.00
d.	Phone	117.42	382.58
e.	Transportation		200.00
f.	Labor		5,000.00
g.	Barber-Nichols		250.00
h.	Martini Engineering Reports		75.00
	Sub-Totals	$ 8,000.00	$ 8,223.58
	Total		$16,223.58

BARBER-NICHOLS
ENGINEERING

FEASIBILITY STUDY OF A RANKINE ENGINE/WOOD COMBUSTOR COGENERATION SYSTEM

October, 1983

Prepared for:

Robert Boucher
721 Desta
Hamilton, MT 59840

Prepared by:

Barber-Nichols Engr. Co.
6325 W. 55th Ave.
Arvada, CO 80002

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1.0 SUMMARY

A preliminary design study of a Rankine engine adapted to a wood combustor for the purpose of cogeneration was made to evaluate the system's expected performance and costs in this application. The purpose of the cogeneration system is to provide electrical power and hot water for heating loads to a small motel or commercial building. The combustor used in this study was a proposed design of Robert Boucher of 500,000 Btu/hr.

Performance calculations indicate that at design thermal input conditions to the engine, the expected power output would be 10.3 kWe for the cycle and system selected which uses steam as the working fluid. Approximately 230,500 Btu/hr would be rejected in the engine's condenser for the heating loads.

The estimated cost for a prototype engine, including design, fabrication and testing, is $255,070. When looking at low quantity production, it is estimated that for 100 units per year the cost could be $62,000 per unit and at 1,000 units per year could be reduced further to $20,000 or less per unit.

2.0 INTRODUCTION

Barber-Nichols performed an engineering study to determine the feasibility of applying a Rankine cycle engine to a wood combustor for the purpose of cogeneration. The cogeneration system would consist of a power module which utilizes a Rankine-type thermodynamic cycle to produce useful electric power, a wood-fired combustor serving as the heat source and a storage tank for storing heat rejected from the power module. The study centered on preparing a preliminary design of the Rankine engine power module. In addition to illustrating the technical features of the Rankine engine, the preliminary design was used to estimate the cost of prototype and production units. This information is to be used by Robert Boucher to determine the economic feasibility of a cogeneration system.

3.0 ENGINE DESIGN

The preliminary engine design evolved in three major phases. Candidate working fluids were initially screened for best potential performance. Next, component specifications were prepared for the top performing fluids and preliminary cost information collected. Finally, the optimum fluid was selected and a control philosophy conceptualized to complete the engine design.

3.1 CYCLE ANALYSIS AND FLUID SELECTION

Candidate working fluids, including refrigerants R-113, R-114 and R-115, the hydrocarbon toluene and steam, were screened for potential performance by computer-modeling the basic Rankine cycle with the above fluids. The basic system is shown on the schematic of Figure 1a. It consists of a high temperature heat exchanger that transfers heat energy from the hot combustion gases to the working fluid which is at relatively high pressure. The working fluid is expanded in a turbine where shaft power is extracted. The working fluid leaving the turbine is still a vapor but at a lower temperature and pressure. The working fluid then flows to the condenser where heat is rejected to cooling water and working fluid condensation takes place. The condensed working fluid is pumped back to the high temperature heat exchanger, thus completing the cycle.

A common variation on this basic cycle is the addition of an internal heat exchanger, commonly called a regenerator (Figure 1b). The regenerator transfers the sensible heat of the turbine exhaust stream to the cold working fluid leaving the condenser. This effectively reduces the amount of heat added to the high temperature heat exchanger and increases the engine efficiency. Since the regenerator represents additional equipment costs, it is included only when economically justified.

The heat source characteristics and the Rankine cycle parameters for the initial fluid screening are shown in Table I. It should be noted that while 60 lbs of wood/hr may have a heating value of 500,000 Btu/hr as noted in Table II, the heat available to the engine, based on the combustion gas conditions of 575 lb/hr mass flow rate and an 1,800°F gas temperature, is approximately 400,000 Btu/hr assuming the gases are cooled to 70°F and 95% of moisture is condensed. Since the Rankine engine is intended as a topping cycle, a rejection temperature of 160°F was set for the cooling water leaving the condenser. Therefore, the Rankine cycle working fluid will be condensing at a temperature greater than 160°F. An assumed condensing temperature of 175°F was selected for initial fluid screening. Since the working fluid leaving the condenser and entering the preheater will be approximately condensing temperature, or higher in a cycle with a regenerator, the combustion gases leaving the preheater will be at some temperature above 175°F depending on the minimum approach temperature for the heat exchanger. Since the dew point in the combustion gases is 148°F, no condensation will occur in the preheater. Therefore, the latent heat contained in the water vapor and a small amount of sensible heat is also not available to the engine. Based on the conditions stated in Table II, approximately 276,000 Btu/hr is the design thermal input to the engine.

Of the preliminary fluids screened, toluene was dropped from consideration because of the potential safety problems associated with its flammability. Toluene would be a very attractive working fluid for this application, as it could provide 45% more power. The flammability aspect can be handled but the problem arises in getting standards or requirements established for such an installation from the various regulatory or code-setting agencies. R-115 was not a practical choice since its critical temperature is 175.6°F, approximately the selected condensing temperature. Of the remaining choices, R-113 and steam proved to be the most promising and were selected for further evaluation.

The potential electrical output for these two fluids at the screening cycle conditions are shown in Figures 2 and 3. The turbine and pump efficiencies for the steam cycle are lower than for the other cycles, as noted on the figure, to reflect realistic performance with the extremely small flow rates at the design point conditions. It was decided to increase the condensing pressure for the steam cycle to 15 psig so that the engine condenser would operate at or above atmospheric pressure. The results of this analysis are shown in Figure 4. Although increasing the condensing pressure reduces output for a constant steam temperature and pressure, operating the engine above atmospheric pressure eliminates a noncondensible gas purge system and simplifies system operation.

It can be seen by comparing Figures 3 and 4 that, despite the lower component efficiencies of the steam cycle, the potential output of the steam cycle is greater than for the R-113 cycle. This is a result of the higher cycle efficiencies obtainable when operating at higher temperatures. The R-113 cycle is limited to 425°F to avoid thermal degradation of the working fluid.

At 750°F and 750 psia, a steam cycle condensing at 15 psia has the potential of generating 11.5 kWe. This steam cycle was selected for further evaluation even though higher temperatures and pressures produce more power. Higher temperature and pressure require more expensive heat exchanger construction materials and could cause overheating of the alternator which is directly shaft-coupled to the turbine. A 425°F superheated, 361.6 psia, R-113 cycle with a net output potential of 9.8 kWe was also further evaluated.

Turbine and pump efficiencies were calculated for the actual cycle conditions. The calculated turbine efficiency for the steam cycle was 54%, six points lower than assumed. The turbine efficiency for the R-113 cycle was calculated at 74%, one point less than assumed and the R-113 calculated pump efficiency was 52%, 12 points greater than the assumed value. Taking the component efficiencies into account, the estimated power output of the steam cycle becomes 10.3 kWe compared to 10.1 kWe for the R-113 cycle.

Since both cycles have very similar power output, cost estimates were prepared for both. The steam cycle resulted in a lower cost engine as it does not require a regenerator and uses a less costly working fluid. The R-113 cycle without a regenerator produces 8.2 kWe. A steam cycle was therefore selected for the preliminary engine design and final cost estimate.

3.2 DESIGN CYCLE

The engine schematic (Figure 5) combined with the listing of thermodynamic state points (Table II) shows the fluid conditions at various locations in the engine. Superheater vapor flows through the turbine start/stop valve to the turbine where shaft power is extracted from the steam. The turbine is directly shaft-coupled to the alternator which generates high frequency alternating current power. The exhaust vapor leaving the turbine enters the condenser where it rejects heat to the cooling water and condenses. From here the condensate flows to the hotwell by gravity. The condensate is then pumped to the feed pump inlet with an electric motor-driven boost pump. In the feed pump, the liquid pressure is raised to its maximum cycle pressure of 775 psia. The pressurized liquid then enters the preheater where it gains heat from the combustion gases and exits as a saturated liquid. Vaporization of the water occurs in the drum-type natural-circulation boiler. Finally, the saturated vapor enters the superheater. Here heat removed from the combustion gases superheats the steam to its maximum cycle temperature of 750°F to complete the cycle.

Approximately 269,000 Btu/hr are removed from the combustion gases in the engine's high temperature heat exchangers. The combustion gases enter the superheater at state point 10 with a temperature of 1800°F where they are cooled to 1480°F. In the boiler/preheater, more heat is removed and the combustion gases leave this heat exchanger at 336°F (state point 12), well above the combustion gas dew point. Thus, an additional heat exchanger will be required to condense the moisture out of the combustion gases.

The turbine shaft power delivered for this cycle is 12.2 kWe of which 1.2 kWe will be consumed as losses in the alternator and 0.76 kWe will be required to run the boost and feed pumps. The net electrical power is 10.3 kWe for an overall cycle efficiency of 13%. The power required to run the condenser cooling water pump was not accounted for as this pump would still be required without a Rankine cycle engine.

3.3 CONTROL PHILOSOPHY

The economic merit of a cogeneration system requires an evaluation of the incremental increase in fuel consumption required to produce power to other alternatives for providing that same power. Of the energy input to the engine, the largest portion, approximately 85%, is the heat rejected to the condenser. When electric power is the only useful output, the losses in the condenser are chargeable to the power system. In topping cycles, however, the heat rejected in the condenser is used as thermal energy for domestic hot water and/or space heating. Under this condition, only the incremental fuel needed to produce power is chargeable to the power system. Thus, to minimize the cost of fuel per kilowatt-hour produced, the cogeneration system should respond to the domestic hot water and space heating demands, i.e., all the heat rejected in the condenser should be useful thermal energy. Therefore, the engine has been designed for variable thermal inputs so that the heat rejected in the condenser matches the process heat requirements.

An estimate of the power out of the Rankine engine as a function of thermal input is shown in Figure 6. From this figure one can see that at 34% of the design thermal input (or approximately 92,000 Btu/hr), the engine produces no net power. The engine can be operated at lower thermal inputs. The alternator will be completely unloaded producing no power and the turbine speed will drop off. The parasitic pump power will still be required to pump the working fluid around the loop. An alternative would be to bypass the Rankine cycle when no net power is being produced and use the circulating pump to pump cooling water directly through the high temperature heat exchangers. This approach would result in higher capital cost for the additional pumping and valves required but would eliminate some operational costs when building heat loads are less than 92,000 Btu/hr.

Alternatively, if the system were equipped with a large thermal storage system, it would be possible to run the system at full input, or some moderate level and produce power while bringing the storage temperature up to a high limit. The storage then would provide the source for the low loads until its temperature dropped low enough for the system to be cycled on again. If combustion efficiency is better at the higher heating rates, this mode of generating heat with off-on cycling may be more efficient than low level modulation, although it will require a larger, more expensive storage system.

There are many ways to configure the interface between the R/C engine and the heating system for the installation. The approach selected will affect the controlability of the system and probably the overall efficiency. The system shown in Figure 5 is one way of interfacing the two functions that appears to be feasible and provide the required control but there may be better ways or additional refinement that improves controlability and overall system operating efficiency. This interface and the control aspects should be evaluated in more detail before selecting a design approach.

4.0 COMPONENT DESCRIPTION

The major engine components include the heat exchangers, rotating machinery to convert thermal energy into electrical power and to pump the working fluid and a control system for the startup and operation of the engine.

4.1 HEAT EXCHANGERS

The heat exchanger employed in this design include a preheater, boiler, superheater and a condenser. A summary of heat exchanger specifications is presented in Table III.

A preliminary preheater/boiler design is shown in Figure 7. The preheater/boiler, a drum-type natural circulation heat exchanger, consists of two rows of 21 parallel, finned, tubed risers. The tubes, 1/2 inch in diameter, and fins are of carbon steel construction as the gas temperature entering this section is less than 1500°F. Natural circulation is developed by the density differential between full water in the downcomers and the steam/water mixture in the risers. The steam and water separate in the drum where the steam is taken off at the top and flows to the superheater.

The superheater, located upstream of the preheater/boiler, is a finned coil constructed of stainless steel due to the 1800°F inlet combustion gas temperature. The superheater is designed to provide the proper amount of superheat at the design conditions. At higher thermal inputs, the steam will be slightly more superheated while at lower thermal inputs the temperature will be below design values.

The water-cooled condenser is a standard tube-and-shell heat exchanger commercially available. The condenser design will permit noncondensible gases that collect in the condenser to be vented to the atmosphere while the system is operating.

4.2 ROTATING MACHINERY

A single-stage, axial flow turbine was chosen for the application. The design specifications are summarized in Table IV. The wheel would have a 5.497 inch pitch diameter and operates at a shaft speed of 60,000 rpm. At the design point, the calculated turbine efficiency is 54%. A multi-stage turbine would provide higher efficiency but the high costs attendant with a multi-stage design generally make them impractical for a low power system such as this one.

The unique feature of the engine is its turbo-alternator power unit. The turbine is directly shaft-coupled to a permanent magnet alternator (PMA) which generates power at a frequency dependent upon shaft speed. Because of the 60,000 rpm shaft speed, the frequency will be much greater than 60 Hz. The alternator output will be rectified to direct current and then inverted to produce the desired 60 Hz power. The turbo-alternator design provides a totally closed, hermetic system.

The feed pump is a commercially available triplex plunger pump. The maximum head differential required is about 2000 feet at a flow of 0.5 gpm. Special high temperature packings must be used.

4.3 CONTROLS

The engine is designed for variable thermal input in order that the condenser heat load matches the required building heat load. As the building heat load varies, the fuel feed rate to the combustor is correspondingly adjusted. This results in variable thermal input to the engine. As the heat supplied to the engine varies, a corresponding change occurs in boiler pressure. With fixed turbine nozzles and a pressure ratio across the nozzles high enough to insure choked flow, the steam mass flow rate will change with heat input in response to the changing boiler pressure. To control feed water level in the boiler drum in response to the flow rate through the turbine, a high pressure float valve is used. As the level in the drum decreases, the float causes the valve to open to allow more flow into the drum. The remaining flow from the positive displacement pump is bypassed back to the hotwell.

The turbine/alternator assembly is designed for maximum performance at variable thermal input. As the heat input varies, thus varying the turbine inlet pressure, the turbine speed will be varied by adjusting the alternator load so that the turbine efficiency is always optimum for the operating turbine pressure ratio.

Make-up water will be supplied to the system at the hotwell in response to a float valve in the hotwell. In order to prevent water contamination which could result in equipment corrosion problems, it is recommended that demineralized and degassed water be pumped into the hotwell when make-up water is required. A leak-tight system should keep make-up water requirements at a minimum.

The steam cycle has been designed to operate the condenser above atmospheric to eliminate a continuous purge system. To prevent low condensing pressures when the system is operating at partial thermal input loads, a temperature-operated throttling valve will sense temperature at the condenser exit on the coolant side and decrease coolant flow rate when the temperature drops below the set temperature.

A heat purge unit has been placed in the condenser coolant loop to prevent condenser overpressure in normal operating modes. Although the system will operate at partial thermal input loads in response to heating load requirements, the heat rejected in the condenser will not always match the rate heat is being utilized from the storage tank on a continuous basis. In the event the temperature of the cooling water from the storage tank becomes too high, the switch will turn on the fan and heat will be rejected to the air.

In addition to the controls necessary for normal operation at design and partial thermal input loads, the engine is supplied with safety controls in the event of system failure. Safety equipment will include boiler relief valves, condenser overpressure and overtemperature switches, safety shutdowns to prevent the engine from damage due to the loss of critical items required for operation such as loss of condenser water flow, loss of bearing lubrication, condenser overpressure and boiler overpressure, loss of alternator load and turbine overspeed.

5.0 COST ESTIMATE

The design concept presented in the preceeding sections was used to estimate the cost to design and fabricate a prototype and production units. The costs reflect those associated with the R/C engine components, its generator and AC converter only. No attempt was made to establish costs on the combustor, the hot gas duct work, flue gas condenser, flue gas blower or the system components associated with the process heating water loop.

The design presented in this report is preliminary in nature and more engineering work is required to integrate the R/C engine with the stove or furnace and to integrate it with the heating syys-tem. The configuration of the heating loop and the R/C condenser needs to be evaluated to ascertain the best approach for meeting heating needs, R/C condensing requirements and best overall performance. The approach selected should consider the benefits of short runs at full power versus partial power modulation. Although this preliminary work is required to develop firm design specifications for the R/C engine and the total system, the costs estimated and presented here for the final design, fabrication and testing of the engines are not expected to change.

5.1 ENGINEERING PROTOTYPE

The following estimated costs are based on having a complete set of design and performance specifications developed and that there aren't any major development problems. The budgets presented do not include the costs associated with extensive formal reporting and documentation of such a program.

Final Design

Includes final layout, detail design, assembly drawings and parts lists.

Labor	$ 63,000
Material & Subcontracts	15,000
Total	$ 78,000

Fabrication and Assembly

Includes purchase of components and material, fabrication and assembly on a skid ready for testing.

Labor	$ 55,320
Materials & Subcontracts	53,250
Total	$108,570

Testing

Includes setting unit up with appropriate simulated loads, calibrating performance, testing for integrity and preparing for delivery. The costs are based on a three-month test period which should ready the unit for application testing.

Labor	$ 58,500
Material	10,000
Total	$ 68,500
Total cost of engineering prototype	$255,070

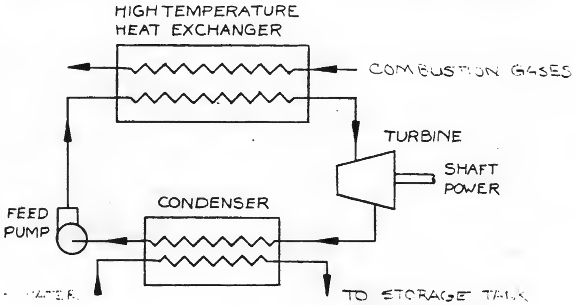
5.2 LIMITED PRODUCTION - 100 UNITS

Production costs are going to be affected by the amount of engineering and tooling invested in achieving producability. Assuming that very little special tooling is designed or made, it is estimated that the engines can be fabricated for $62,000 each. It is felt that these costs could be reduced if it was planned to produce 100 units per year over several years.

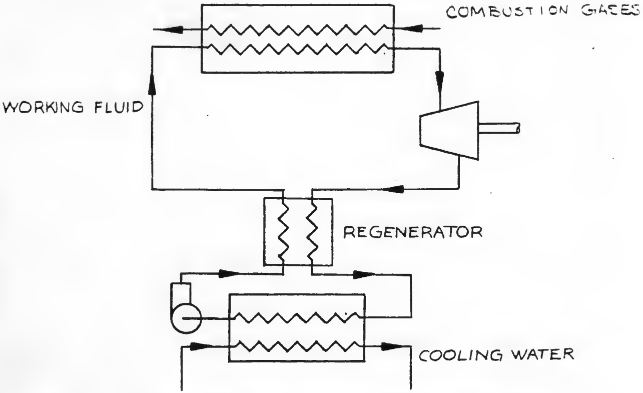
5.2 QUANTITY PRODUCTION - 1000 UNITS

For large quantity production, many factors come into play. To achieve the best production costs, the engine and its components would have to be redesigned for production with appropriate tooling. We are not capable of providing a good estimate of production costs, but if one looks at the costs of a centrifugal water chiller A/C unit, it would appear that an R/C engine of this size could be manufactured in a cost range of 10 to 20 thousand dollars to the consumer. This estimate assumes that the A/C unit is relatively equivalent to the R/C engine in complexity at the same power level.

FIGURE 1 RANKINE CYCLE SCHEMATIC


HIGH TEMPERATURE HEAT EXCHANGER
COMBUSTION GASES
TURBINE
SHAFT POWER
CONDENSER
FEED PUMP
WATER
TO STORAGE TANK FOR HOT WATER

a) BASIC RANKINE CYCLE



b) RANKINE CYCLE WITH REGENERATOR

BARBER-NICHOLS ENGINEERING CO.

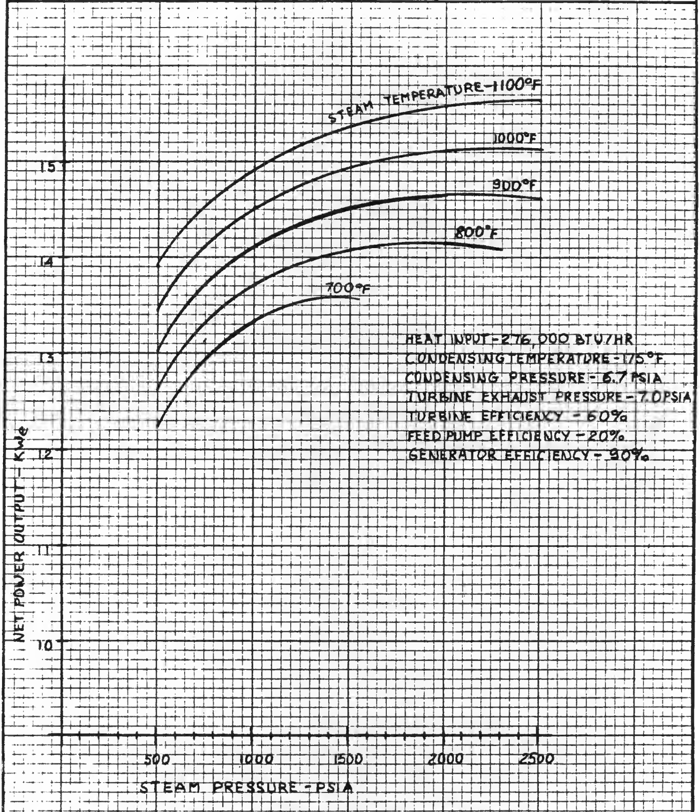
PROJECT

BY

DATE

SUBJECT

INITIAL FLUID SCREENING



HEAT INPUT - 276,000 BTU/HR
CONDENSING TEMPERATURE - 175°F
CONDENSING PRESSURE - 6.7 PSIA
TURBINE EXHAUST PRESSURE - 7.0 PSIA
TURBINE EFFICIENCY - 60%
FEED PUMP EFFICIENCY - 20%
GENERATOR EFFICIENCY - 90%

RANKINE ENGINE GENERATING POTENTIAL
FIGURE 2

BARBER-NICHOLS ENGINEERING CO.

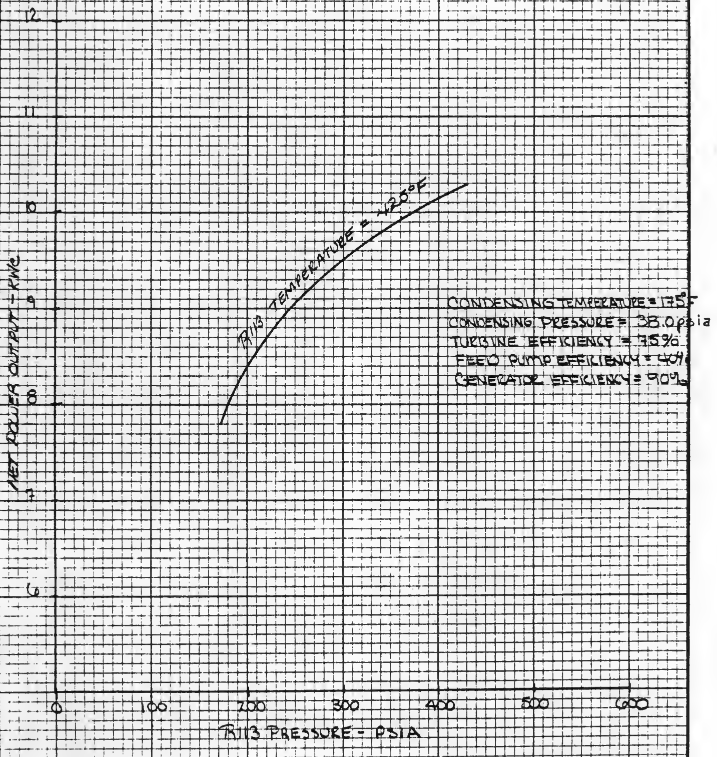
PROJECT

BY

DATE

SUBJECT

INITIAL FLUID SCREENING



CONDENSING TEMPERATURE = 175°F
CONDENSING PRESSURE = 38.0 psia
TURBINE EFFICIENCY = 75%
FEED PUMP EFFICIENCY = 40%
GENERATOR EFFICIENCY = 90%

RANKINE ENGINE GENERATING POTENTIAL

FIGURE 3

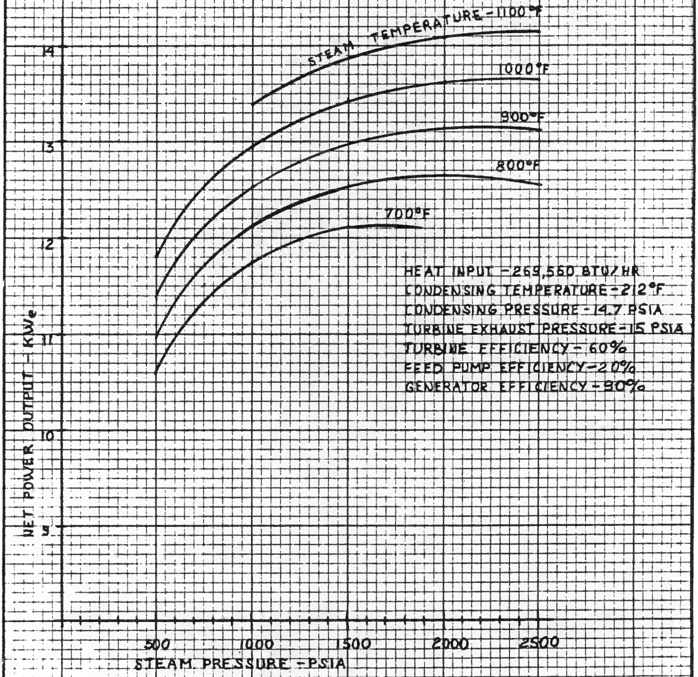
PROJECT

BY

DATE

SUBJECT

INITIAL FLUID SCREENING



HEAT INPUT - 269,560 BTU/HR
CONDENSING TEMPERATURE - 212°F
CONDENSING PRESSURE - 14.7 PSIA
TURBINE EXHAUST PRESSURE - 15 PSIA
TURBINE EFFICIENCY - 60%
FEED PUMP EFFICIENCY - 20%
GENERATOR EFFICIENCY - 90%

RANKINE ENGINE GENERATING POTENTIAL

FIGURE 4

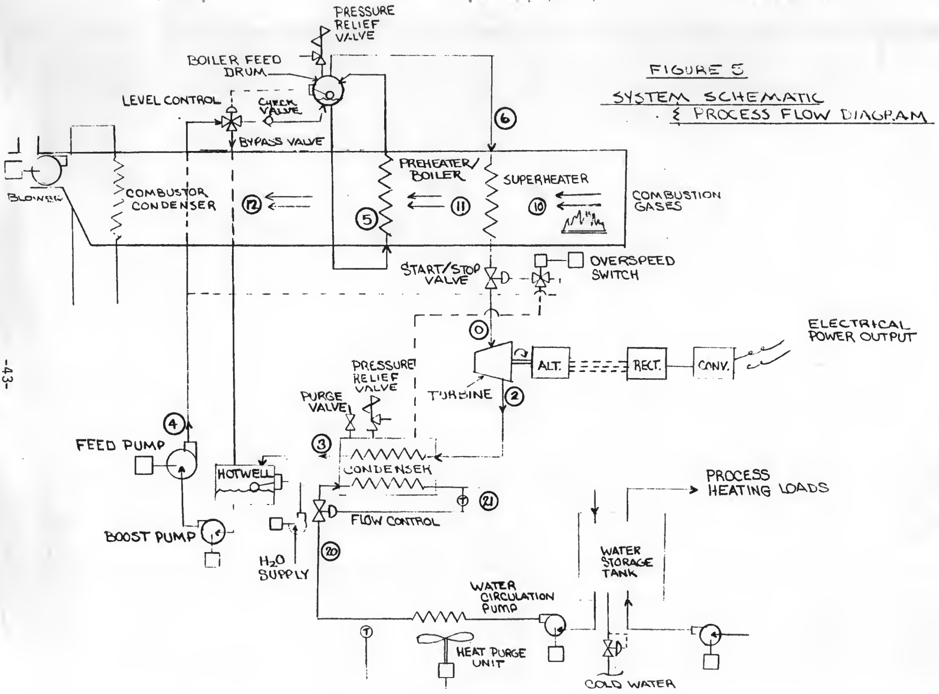
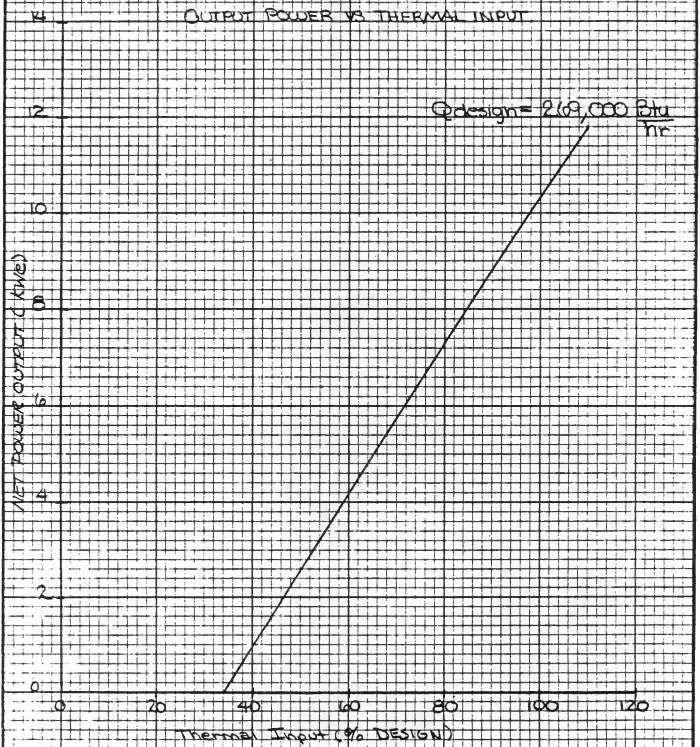


FIGURE 5

SYSTEM SCHEMATIC & PROCESS FLOW DIAGRAM

BARBER-NICHOLS ENGINEERING CO.

PROJECT MS1310		SUBJECT Off - Design Performance
BY Cind, C	DATE 9/14/82	



OFF - DESIGN PERFORMANCE
FIGURE 6

BARBER-NICHOLS ENGINEERING CO.

PROJECT		SUBJECT
MS-1310		boiler- preheater
BY SO	DATE 9-16-83	

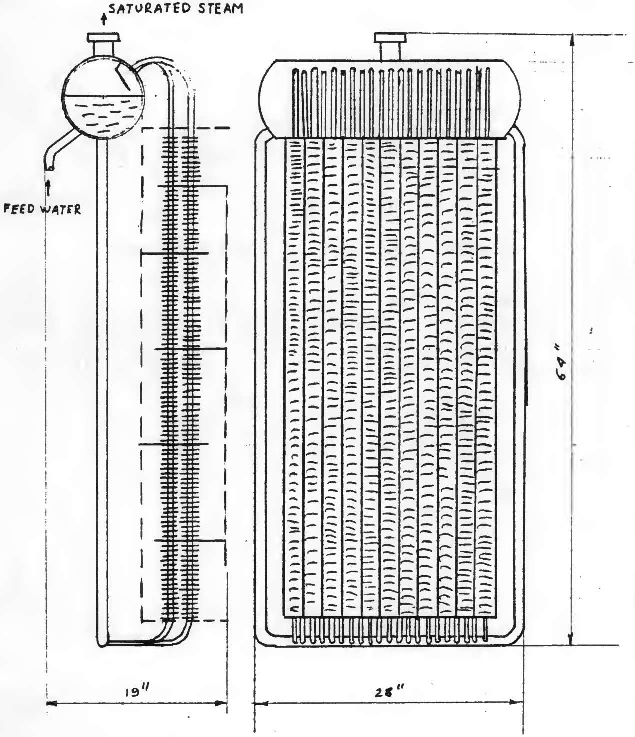


FIGURE 7

PRELIMINARY BOILER DESIGN

TABLE I

CHARACTERISTICS OF THE BOUCHER WOOD COMBUSTOR

Heat Input = 500,000 Btu/hr = 60 lbs wood/hr

Combustion Gas Data:

Gas Flow Rate = 575 lbs/hr
Gas Temperature = 1800°F (measured)
Gas Analysis (weight %) = 18.8% CO_2, 15.6% H_2O, 52% N_z, 13.6% Excess Air
Gas Specific Heat = .32 Btu/lb-°F

RANKINE CYCLE PARAMETERS USED FOR INITIAL FLUID SCREENING

Condensing Temperature: 175°F
Minimum Heat Exchanger Approach Temperature Difference: 120°F
Regenerator Effectiveness: 0.80
Working Fluid Pressures Drops:
  Feed Pump to Boiler Exit: 20 psid
  Boiler Exit to Turbine Inlet: 5 psid
  Turbine to Condenser: 0.5 psid
Component Efficiencies:
  Alternator: 90%
  Feed Pump and Drive: 40%
  Turbine: 70%

TABLE II
CYCLE THERMODYNAMIC STATE POINTS

State Point	Pressure (psia)	Temperature (°F)	Enthalpy (Btu/lbm)	Density (lbm/ft^3)
Steam				
0 - Turbine Inlet	750	750	1372.5	1.120
2 - Turbine Exit	15	295	1190.2	0.033
3 - Condenser Exit	15	212	181.2	59.8
4 - Feed Pump Exit	775	223	192.6	59.7
5 - Preheater Exit	755	511	501.1	48.3
6 - Boiler Exit	755	511	1200.7	1.640
Flow Rate (lb/hr) = 228.5				
Combustion Gases				
10 - Superheater Inlet	-	1800	-	-
11 - Preheater/ Boiler Inlet	-	1587	-	-
12 - Combustor Condenser Inlet	-	336	-	-
Flow Rate (lb/hr) = 575				
Cooling Water				
20 - Condenser Inlet	-	140	-	-
21 - Condenser Outlet	-	160	-	-
Flow Rate (lb/hr) = 11,522				

TABLE II (Continued)

Overall Engine Parameters

Turbine Efficiency = 54%
Alternator Efficiency = 90%
Pump/Motor Efficiency = 20%

Heat In = 79.0 kW_{th}
Heat Out = 68.7 kW_{th}
Net Power Out = 10.3 kWe
Cycle Efficiency = 13.0%

TABLE III
HEAT EXCHANGER SPECIFICATION SUMMARY

HIGH TEMPERATURE HEAT EXCHANGERS

Combustion Gas Flowrate (lbs/hr) = 575
Steam Flowrate (lbs/hr) = 210

Hot Side (Combustion Gases)	Superheater	Boiler	Preheater
Temperature In (°F)	1800	1587	715
Temperature Out (°F)	1587	715	336
Cold Side (Steam)			
Temperature In (°F)	511	511 (liq)	223
Temperature Out (°F)	750	511 (vap)	511
LMTD (°F)	1063	524	154
Overall Heat Transfer Coefficient (Btu/hr-ft^2-°F)	7.5	8.0	8.0
Heat Transfer Area (ft^2)	4.9	38.1	57.2
Total Heat Transferred (Btu/hr)	39,300	159,800	70,500

CONDENSER

Hot Side
Steam Flowrate (lb/hr) = 228.5
Temperature In (°F) = 295 (vapor)
Temperature Out (°F) = 212 (liquid)

Cold Side
Water Flowrate (lb/hr = 11,522
Temperature In (°F) = 140
Temperature Out (°F) = 160

MTD (°F) = 61.5
Overall Heat transfer Coefficient (Btu/hr-ft^2-°F) = 250
Heat Transfer Area (ft^2) = 15
Total Heat transferred (Btu/hr) - 230,506

TABLE IV
TURBINE CONDITION AT DESIGN POINT

Turbine Inlet Pressure (psia)	750
Inlet Temperature (^{O}F)	750
Exhaust Pressure (psia)	15
Mass Flowrate (lb/hr)	228
Power Output (hp)	16.28
Shaft Speed (rpm)	60,000
Specific Speed (CFS Units)	7.16
Pitch Diameter (in.)	5.497
Blade Number	128
Blade Height (in.)	0.3135
Nozzles	1
Efficiency (%)	53.81

$$\frac{\mathrm{RPM}\ (\mathrm{CFS})^{\frac{1}{2}}}{\mathrm{ft}^{3/4}}$$

BARBER-NICHOLS ENGINEERING

November 14, 1983

Bob Boucher
Star Route Potomac
Bonner, MT 59823

Dear Bob:

Enclosed is the final report on our evaluation of a Rankine cycle engine applied to your wood burning stove. Technically it would be a good application with an overall efficiency of 13% while providing 160°F water for heating needs. The Rankine cycle engine does not drop the flue gas temperature far enough to condense the moisture present and clean up the gases. There is considerable heat energy available in this gas, but no attempt was made to utilize it or integrate it into the system because of the low temperatures required on your exit gas stream. It would appear that part of the energy could be used if the separate gas cooler and condenser were designed for a high temperature rise. If the cooling water for this heat exchanger can exit at temperatures above those needed for the heating system, then part of the energy can be used. An external cooler would then be required to drop the cooling water temperature enough to obtain the required exhaust gas temperature.

The prototype costs are reasonable based on past programs with similar equipment. Additional engineering work needs to be done to definitize the engine interfaces to the heating system and to the stove so that a firm, complete set of specifications can be developed for the engine. I estimate that the work needed to

6325 West 55th Avenue, Arvada, Colorado 80002 Telephone 303-421-8111

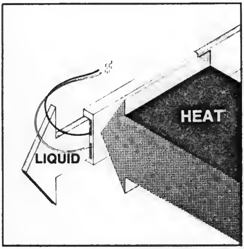
develop the final specifications would cost between $10,000 and $15,000.

We appreciate the opportunity to evaluate the application of a Rankine cycle engine to your stove. I hope that it is economically justifiable and that we can pursue the project further with you. If you have any questions or need further information, please call.

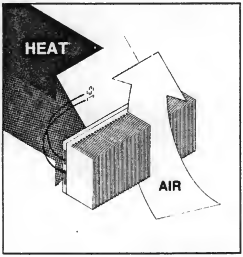
Sincerely,
BARBER-NICHOLS ENGR. CO.
Bob Olander
Robert G. Olander

RGO/dsb

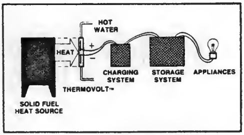
Encl.



LIQUID SYSTEM



AIR SYSTEM



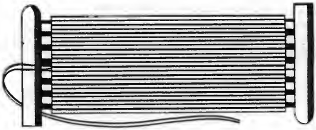
REMOTE POWER

Thermoelectrics, thermal generators, or reverse Peltier effect are all terms used to describe the direct conversion of heat to electricity. We at Photic Corporation refer to it as thermovoltaics. P-and N-junction semiconductors with thermovoltaic properties are kept cool on one side and hot on the other. This temperature differential across the thermovoltaic semiconductors causes voltage to develop and current to flow. *Electricity from heat.*

The Thermovolt™ is designed to be used in locations where renewable solid fuels, like wood, are readily available and a heat source is desireable. Therefore, electricity is stored not only in your battery bank, but also in your wood pile. Unaffected by adverse weather conditions, the Thermovolt™ system also provides heat for space heating and domestic hot water. Thermovolt™ thermovoltaic systems can compete favorably with other energy conversion technologies due to its more constant and controllable form of electrical generation. Storage systems can be scaled down because of this more continuous generation.

The Thermovolt™ system provides a reliable power source for lighting, refrigeration, and communications, all accomplished with few or no moving parts. Thermovoltaics are here and the possibilities are endless.

Thermovolt™ charging the way, today.



OUTPUT: 720-1200 Net Watt-hours/day with the 6"x 14" Thermovolt™ standard liquid module.

Photic Corporation is an energy conversion technology firm located in Traverse City, Michigan, USA. Photic Corporation also manufactures Heliopass™ solar collectors and Finductor™ heat exchangers.

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Sundstrand Energy Systems
unit of Sundstrand Corporation

4747 HARRISON AVENUE, ROCKFORD, IL 61101 • PHONE 815 226-6000 • TWX 910 631-4255 • TELEX 25-7440

ESD3-5-4615LM
ORC 1538
11 May 1983

Mr. Bob Boucher
Star Potomic
Bonner, MT 59823

Dear Mr. Boucher:

Enclosed is the information describing Sundstrand's waste heat recovery system which we discussed. This system could be applied to a wood burning furnace. However, the unit is rated at 750 kw and it would not be economically suitable for an application requiring only 50 to 100 kw.

Very truly yours,

SUNDSTRAND ENERGY SYSTEMS
A UNIT OF SUNDSTRAND CORPORATION

Doug Lacey

Doug Lacey
Product Line Manager
Commercial Energy Systems

DL/bn
Encl.

POWER GENERATION FROM WASTE HEAT USING ORGANIC RANKINE CYCLE SYSTEMS

Anil Prasad
Product Manager
Sundstrand Energy Systems

POWER GENERATION FROM WASTE HEAT USING ORGANIC RANKINE CYCLE SYSTEMS

Anil Prasad
Product Manager
Sundstrand Energy Systems

ABSTRACT

Many efforts are currently being pursued to develop and implement new energy technologies aimed at meeting our national energy goals. The use of organic Rankine cycle engines to generate power from waste heat provides a near term means to greatly increase the fuel utilization efficiency of many processes. Sundstrand has developed an organic Rankine cycle unit to recover energy from various waste heat sources. A field test program was jointly funded by Sundstrand and the U.S. Department of Energy. As part of this program, four units were placed in the field to demonstrate the economic viability and the reliability of the equipment in commercial and industrial applications. This paper describes the system and the field test program.

INTRODUCTION

The Sundstrand Organic Rankine Cycle (ORC) Waste Heat Power Conversion System is intended to be a universal bottoming cycle that can convert the energy in waste heat streams into usable shaft power. The nominal rating of the unit is 750 KWe or 1,100 SHP. The basic bottoming cycle concept is shown in Figure 1.

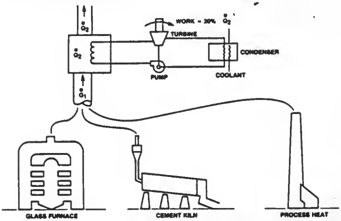


Figure 1. Bottoming Cycle Concept

By changing the configuration of the waste heat recovery vaporizer, the unit can accommodate any gaseous waste heat stream that has a temperature above 600°F and sufficient flow. The vaporizer can also be designed for liquid or condensing waste heat streams with temperatures above 500°F, which makes it well suited for solar power systems using concentrating collectors.

SYSTEM OPERATION

The system operates on Rankine cycle which is shown in schematic form in Figure 2. The diverter valve directs the waste heat stream through the vaporizer. The working fluid is boiled in the vaporizer. The saturated vapor expands through the turbine, generating mechanical power. This expansion process causes the working fluid to become highly superheated. The vapor is desuperheated by passing it through a regenerator, then condensed and collected in a reservoir. The pumping system increases the working fluid pressure and sends it to the regenerator for preheating. The preheated liquid then flows to the vaporizer, thus completing the cycle.

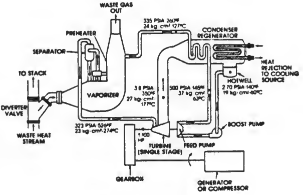


Figure 2. ORC Schematic

The cycle as shown in Figure 2 along with the working fluid state points is used for system operation with heat source temperatures above 750°F. A second version of the system that uses the same basic hardware but operates with a turbine inlet temperature of 484°F at 230 psi is applied to heat streams above 600°F. The design cycle efficiencies are 22.1% and 20.9%, respectively.

The unit is designed for unattended operation. In suitable applications even the startup and shutdown can be performed totally automatically.

SYSTEM DESCRIPTION

The system is a closed organic Rankine cycle using as many off-the-shelf components as possible, creating a state-of-the-art design. The system is a complete package consisting of all the

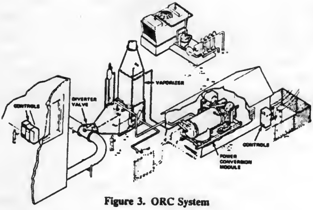


Figure 3. ORC System

equipment required for startup and unattended operation with the exception of the cooling tower. The equipment supplied by Sundstrand is shown in Figure 3 and consists of the following: 1) diverter valve, 2) vaporizer, 3) power conversion module (PCM), and 4) controls.

Diverter Valve

The diverter valve is a three-way, pneumatically actuated valve that controls the flow of the waste heat stream through the vaporizer. During normal operation of the ORC unit, the diverter valve is an active control element, regulating the waste heat flow through the vaporizer, thus maintaining the proper turbine inlet temperature. When the ORC unit is shut down, the diverter valve routes the waste heat stream through the normal exhaust gas path. The valve design is dependent on application and can be a single package or two separate valves.

Vaporizer

The external configuration and internal design of this component are dependent on the application but will generally appear as shown in Figure 4. The vaporizer contains a steel finned tube preheater and boiler of similar construction. The preheater has a multiple pass serpentine counterflow arrangement and is located in the vertical section. The boiler is of a natural circulative design located in the horizontal section. The working fluid exits the boiler as saturated vapor and enters a centrifugal separator that allows vapor flow to the turbine while recycling the liquid to the boiler. The heat addition to the working fluid will be 13.4 x 10^6 Btu/hr for a nominal 200 psi system or 12.7 x 10^6 Btu/hr for a nominal 300 psi system.

The component design, manufacture, and stamping is in accordance with Section I and Section VIII of the ASME Boiler and Pressure Vessel Code.

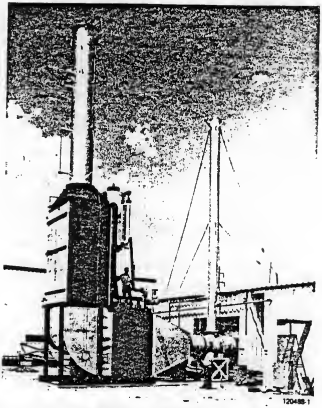


Figure 4. Standard Vaporizer

Power Conversion Module (PCM)

As the component title implies, this component recovers the energy from the waste heat stream and converts it to usable mechanical power than can be used directly or converted to electricity. The standard PCM, as shown in Figure 5, contains a supersonic impulse turbine with input operating conditions of 230 psi, 484°F or 323 psi, 524°F. These conditions rotate the output shaft at 9,300 or 10,100 rpm respectively, generating 1,100 shaft horsepower. A gearbox is employed to slow the shaft to whatever speed is appropriate for the driven equipment.

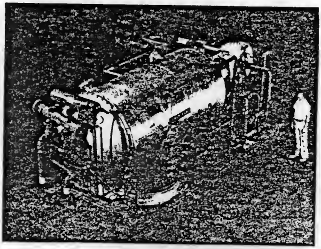


Figure 5. Power Conversion Module

The PCM is also equipped with the condenser/regenerator housing that contains the condenser and regenerator heat exchangers. These heat exchangers de-superheat and convert the working fluid vapor to a liquid that is collected in the reservoir. The housing and heat exchangers are designed to Section VIII of the ASME Boiler and Pressure Vessel Code.

The pumping system of the ORC engine is also within the boundaries of the PCM. The working fluid is drawn from the reservoir by a boost pump that partially pressurizes the fluid. The final pressurization of the fluid is accomplished by the feed pump. The feed pump is a two stage, back-to-back, vaned impeller contained within the same housing as the turbine and connected directly to the turbine shaft. This combined rotating unit is supported by working fluid lubricated fluid film bearings.

Controls

The control system, as shown in Figure 6, includes the controls for the ORC system as well as the controls for the driven equipment. These allow the system to operate unattended in two modes, infinite bus and finite bus fixed speed.

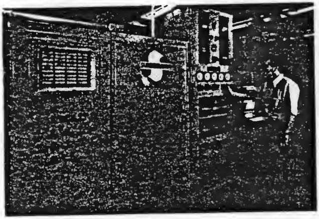


Figure 6. Controls

Working Fluid

The fluid selection criteria were toxicity, availability, resulting system cost, flammability, and performance; listed in order of importance. Low toxicity and current industrial acceptance were considered to be very important. Availability was a must and some of the more desirable fluids failed this test. The effect of the working fluid on cost included the cost of the fluid itself, any special materials required for construction, and the thermodynamic effects, e.g., heat exchanger size, number of turbine stages, etc. These effects were included in the cost optimization that was carried out for water and the few organic fluids that survived the screening process. The optimization analysis indicated that toluene was the leading contender among organic working fluids and provided a cost effective system. Steam systems remained considerably more costly because of the large waste heat boiler size and complex turbomachinery. Therefore, the decision was made to use toluene as the working fluid. Sundstrand has had considerable experience with toluene, and in addition to having desirable thermodynamic and physical characteristics, it is inexpensive, widely used, well understood, and of low toxicity.

APPLICATIONS

The Sundstrand ORC unit has been designed to be as flexible as is practical with respect to application. The vaporizer, diverter valves and controls can be readily adapted to a wide variety of heat sources. The driven equipment can be selected to meet the requirements of the user.

One of the initial applications for the ORC system has been at municipal utilities. The exhaust gases from large reciprocating diesel engines are very suitable for the ORC unit. The heat source is generally clean and easily accessible, and the vaporizer can be designed for several inches of water pressure drop without upsetting the engine efficiency. The gas turbine exhausts are equally attractive for use with the ORC system. There are a number of diesel engines and gas turbines operated by municipal utilities and natural gas pipe line compressor stations which are potentially good applications for the ORC system.

Several other applications have been examined where the exhaust from an industrial process is used as the heat source. Each of these applications has its own set of constraints and unique requirements. One characteristic which is common to most of the industrial applications is the need for a draft fan to make up the additional pressure drop caused by the heat recovery boiler and associated ducting. Some of the other conditions which must be examined are minimum temperature to which the waste heat can be cooled without causing fouling and corrosion of the vaporizer and the characteristics of particulate material in the waste heat stream. These properties must be taken into account in the vaporizer design and construction.

Some of the industrial applications with good potential for heat recovery are cement kilns, ceramic kilns, glass furnace, refractory kiln, foundry furnace, fume and solid waste incinerators. In addition, the operating temperatures of the Sundstrand system are such that it is ideally suited for solar power using concentrating collectors.

FIELD TEST PROGRAM

An organic Rankine cycle unit for waste heat recovery from various sources has been developed and built by Sundstrand Energy Systems under a jointly funded program with the U.S. Department of Energy. This cooperative agreement between Sundstrand Energy Systems and the U.S. Department of Energy also provided for installing four field test units at various municipal utilities and industrial sites.

The first development system is installed at Sundstrand in Rockford, Illinois. It is a full size unit operating in parallel with the utility. The unit has accumulated over 8,000 hours and has generated over 4,000,000 KWH.

The field test program was designed to prove the technical and economic viability of the ORC unit for various waste heat recovery applications. The first three of these field test units have been installed in municipal utilities to recover energy from diesel engine exhausts. The fourth unit has been installed at a ceramic kiln.

The four field test sites and units with their performance as of May 1981 are described below briefly.

1) Municipal Power Plant, Beloit, Kansas

The waste heat source is the exhaust from two dual fuel engines. The installation is shown schematically in Figure 7. The generator operates in parallel with the main grid. The unit has accumulated over 8,800 hours of operation and has generated over 4,000,000 KWH.

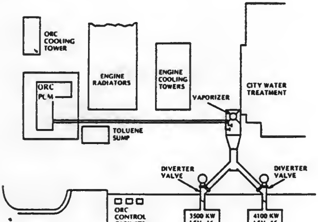


Figure 7. Installation of the ORC unit at Beloit, Kansas, Municipal Utility

The system parameters are as follows:

Heat source

Engine make - Cooper Bessemer
Size (BPH) - 4,910 and 5,744
Exhaust temperature - 821°F

Vaporizer

Pressure drop - 7 in. w.g.
Flow rate - 94,237 lb/hr

Power output

Electrical - 600 KW

2) Municipal Power Plant, Easton, Maryland

The unit is connected to two large diesel engines; however, it can produce full power from the exhaust of one engine only. The installation is shown schematically in Figure 8. The generator is tied to the main grid in parallel. The power plant is operated by remote control from the main power plant located a few miles away. During normal running there are no operators present at the ORC installation. The unit has accumulated over 1,900 hours and has generated over 1,000,000 KWH.

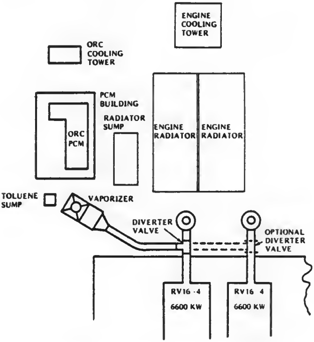


Figure 8. Installation of the ORC unit at Easton, Maryland, Municipal Utility

The system parameters are as follows:

Heat source

Engine make - Delaval
Size (BHP) - 8,658
Exhaust temperature - 752°F

Vaporizer

Pressure drop - 7 in. w.g.
Flow rate - 99,932 lb/hr

Power output

Electrical - 600 KW

3) Municipal Power Plant, Homestead, Florida

This unit is also connected to two large diesel engines as shown in Figure 9. Full power can be developed from one engine exhaust only. The unit has accumulated over 600 hours and has generated over 330,000 KWH.

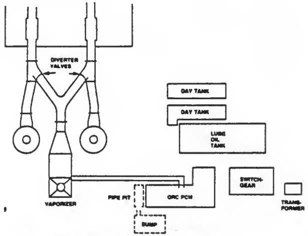


Figure 9. Installation of the ORC unit at Homestead, Florida, Municipal Utility

The system parameters are as follows:

Heat source

Engine make - Delaval
Size (BHP) - 12,216
Exhaust temperature - 752°F

Vaporizer

Pressure drop - 7 in. w.g.
Flow rate - 128,229 lb/hr

Power output

Electrical - 600 KW

4) Crane Company, Ferguson, Kentucky

The ORC unit has been designed to recover heat from ceramic kilns. The installation is shown schematically in Figure 10. The unit is connected to several ceramic kilns. A draft fan has been provided to overcome the pressure drop in the ducts without upsetting the kiln pressure. Each kiln can be individually controlled with dampers and butterfly valves. The generator has been designed to operate in parallel to the utility under normal conditions; however, during utility brownout or blackout the nonessential loads at the ceramic plant are dropped, and the generator switches to stand alone mode to service the essential loads of the kiln.

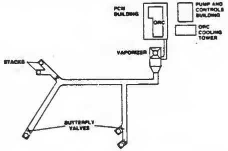


Figure 10. Installation of the ORC unit at Crane Co., Ferguson, Kentucky

The design features are as follows:

Heat source

Ceramic kilns
Exhaust temperature - 781°F

Vaporizer

Pressure drop - 2.4 in. w.g.
Flow rate - 62,010 lb/hr
Waste gas exit temperature - 336°F

I.D. Fan

Supply pressure - 7.5 in. w.g.
Flow rate - 62,010 lb/hr

Power output

Electrical - 680 KW (Max.)

OTHER APPLICATIONS

5) Solar Irrigation

This installation for a solar irrigation application is also DOE funded, but not under the above field test program. The system was installed in November 1979, and is located in Coolidge, Arizona.

The ORC generates 200 KW gross from a modified 600 KW system.

The system parameters are as follows:

Heat source

Heat transfer oil
Temperature - 550°F
Flow - 14,031 lb/hr

Power output

Electrical - 200 KW

Commercial Demonstration Program

Subsequent to the above field test program currently Sundstrand Energy Systems and U.S. DOE are jointly sponsoring a commercial demonstration program. ORC units are available for most industrial applications under this program. The first unit under this program has been sold for a fume incinerator application.

6) Fume Incinerator

This unit is designed for waste heat recovery from a fume incinerator used in a metal decorating operation. The electrical generator is designed to operate in parallel with the utility. The system is scheduled to be started in January 1982. The installation is shown schematically in Figure 11.

The system parameters are as follows:

Flow - 61,200 lb./hr.
Temperature - 838°F
Pressure drop - 6 in. w.g.
Power output - Electrical-440kW

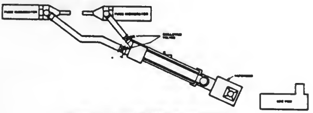


Figure 11. Installation of the ORC unit for fume incinerators.

ECONOMICS

The amount of saving is dependent on several site specific factors and, therefore, each application must be evaluated individually, taking into account specific circumstances. The most significant factors which affect the economics of ORC systems are the cost of electricity, the number of operating hours per year, and the installed cost of the ORC. Figure 12 shows the gross annual savings which will result from the ORC system as a function of energy cost and operating hours per year. Figure 13 shows the after tax return on investment as a function of installed cost per KW and gross annual savings.

The assumptions used in calculating the ROI are:

ORC life - 20 years
Depreciation - 16 years DDB
Investment tax credit - 10%
Income tax rate - 50%
Inflation - 8%
ORC maintenance expense - $.0015/KWH

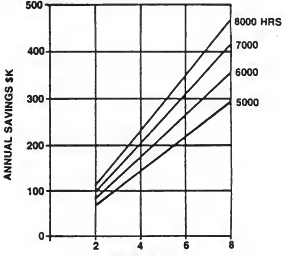


Figure 12. Gross Annual Savings 750 KWe Unit

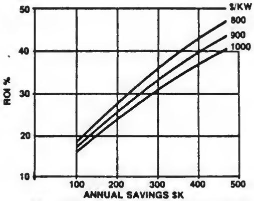
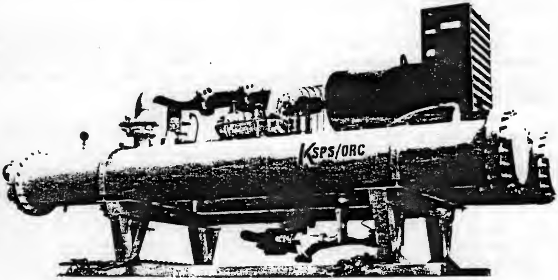


Figure 13. Return on Investment 750 KWe Unit

CONCLUSION

The testing done to date indicates that the ORC design is reliable and exceeds its design performance requirements. As of May, 1981, the field test units together with the development unit have accumulated over 20,000 hours of operation. The field test program plus the solar energy program has provided the necessary reliability and operating cost data to allow introduction of the equipment to the commercial market.

POWER WITHOUT FUEL!



ORC SERIES 75 D/D GENERATOR SET

K S.P.S. INC.

The most efficient means of converting heat into work in use today is the Rankine cycle, used by all major thermal power generating plants. Usually the working fluid employed is water. However, steam has a very low power density at the lesser temperatures encountered in most waste heat sources. Because of its very low molecular weight, a given weight of steam occupies a very large volume. For example, it would require the large volume flow of over six cubic feet per minute of saturated steam at 212°F to generate one horsepower (assuming 90°F condenser temperature).

On the other hand, if a high molecular weight fluid (such as Freon 12) is employed in the same Rankine cycle with the same temperature conditions, the enormously greater density of the vapor would result in a flow rate of less than a third of a cubic foot per minute to generate that same one horsepower.

S.P.S., Inc. is an energy company. Our primary business since 1976 is the manufacture and sale of Organic Rankine cycle engines and generator sets built under US and overseas patents which cover the use of non-volatile liquids in the Rankine cycle, and expander design.

S.P.S./ORC engines may be equipped with a generator (synchronous or induction), an air or gas compressor, or a pump, as required. A power takeoff shaft is provided for ancillary requirements.

Applications for the ORC include (1) Industrial waste heat to power conversion, (2) Solar, and (3) Geothermal. Our experience includes a wide range of heat sources such as low pressure steam exhaust, condensate streams, stack gas, gas turbine exhaust, diesel exhaust and cooling, sawdust burning boilers, cooling systems, solar heated water, and geothermal. The system may be applied to any liquid, vapor or gaseous heat stream from 150°F or higher, provided there is available cooling water at 100°F lower than the heat source. However, in order to keep package cost and site work to a minimum we are not presently accepting contracts for stack reclaimers.

Installation. S.P.S. provides everything except: Concrete pad, Pumps, Valves and Piping of the hot and cold streams, and electrical transformers (if required). Systems over 50 KVA are assembled on site in three days or less at no extra cost.

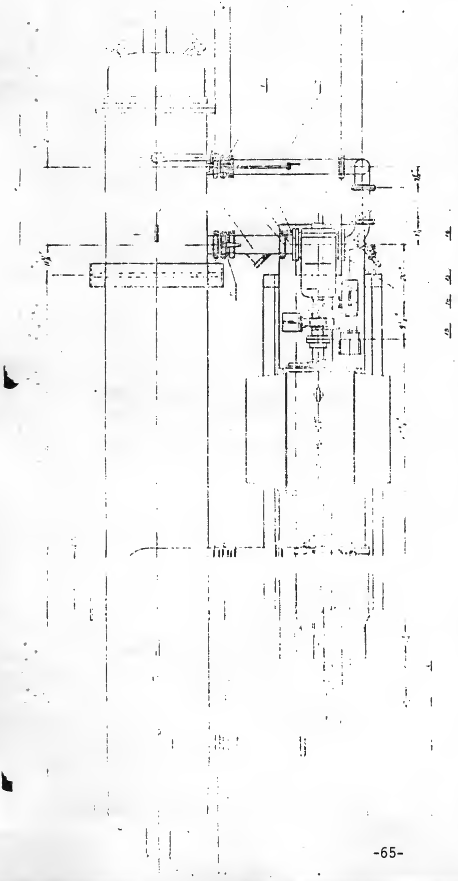
Maintenance. Since the ORC is similar to a commercial air conditioning system running backwards, servicing requirements are identical.

Warranty. All components are protected by standard commercial warranty against defects in workmanship and material for periods of not less than one year. Nominal horsepower is guaranteed when the Freon boiler and condenser are maintained at a 100°F temperature difference. Higher or lower temperature differences will produce proportional power output.

This system is built under U.S. Patents # 3479817, 3636706, 3750393.

*FREON is an E.I. DuPont trade mark.
"SOLAR ENGINE" is an S.P.S., Inc. reg. trade mark.

System Description: All components of S.P.S./ORC systems are built by major American manufacturers. Our suppliers include such companies as Joy, Waukesha, Siemens-Allis, Lima, Bohn, Jamesbury and Victaulic.



Heat Exchangers: Custom designed, code approved shell and tube vessels where Freon is vaporized by application of heat and condensed with a counter current stream of cooling water. Passivation and Cathodic protection is standard. Stainless steel is optional at extra cost.

Freon Pump: Direct coupled positive displacement.

Generator: Induction or synchronous. Voltage, phase and frequency of your choice.

Switchgear: All protective relays required to interface utility are part of standard package.

Expander: Positive displacement rotary screw. More rugged and efficient than a turbine.

Controls: All controls for automatic operation, including automatic speed control are included.

Computation of Recoverable Power. The S.P.S./ORC system converts about 70% of the theoretically available Carnot efficiency into shaft horsepower. Carnot efficiency is dependent upon the temperature difference between the heat stream and the condenser cooling water. In most cases of waste heat streams at temperatures in the range of 150°F - 212°F, from 7% to 15% of the available heat energy is recoverable as shaft horsepower. A higher percentage of energy is recoverable from streams at higher temperature levels. Although the precise calculation of the potential power output from any heat source involves many factors, some simplified equations are provided as follows:

For Hot Water: HP - (TH-Tc) 0.0035 GPM
HP - ORC system shaft output horsepower
TH - Temperature of hot water supply °F
Tc - Temperature of condensing water °F
GPM - Gallons per minute flow of hot water

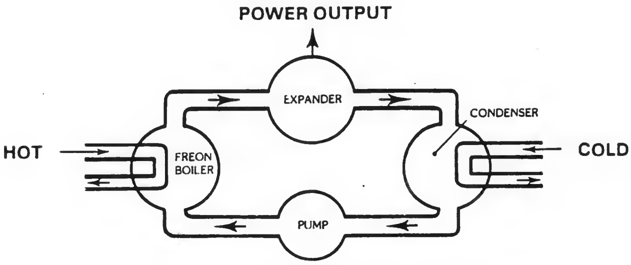
In metric units: HP - (TH-Tc) 0.0017 liters per minute where water temperatures are expressed in °C and flows in liters per minute.

For Steam at Atmospheric Pressure: Where condensing water is in the normal ambient temperature range, substitute available steam flow for the term in parenthesis in the following equation:
HP - 0.038 (lbs steam per hour)
In metric units: HP - 0.084 (Kg steam per hour)

Rule of Thumb: For rough calculation a figure of 35,000 BTU's per HP or 40,000 BTU's per KW may help.

Flow Rates: Vary widely depending on the temperature of the heat stream but as a general rule, the cooling stream will be at least two to three times the volume of the heat stream.

S.P.S. offers a complete range of computer programs which facilitate fast and accurate calculation of recoverable power from customer's heat sources.



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*FREON is an E.I.Dupont trade mark. "SOLAR ENGINE" is an S.P.S., Inc. reg. trade mark.

JUNE 1,1981 (Revised)

KSPS/ORC ENGINE PRICE LIST

HP	KVA OUTPUT	ORC GEN SET FOR HEAT SOURCE OVER 205° F	ORC GEN SET FOR HEAT SOURCE UNDER 205° F	ANNUAL EARNINGS AT 5¢ PER KWH	ANNUAL EARNINGS DIRECT DRIVE	DEMENSIONS L x W x H	SHIPPING WEIGHT IN 1,000 LBS	DELIVERY
14	10	$ 14,391	$ 16,118	$ 4,380	$ 4,905	8'x4'x4'	2.6	90 Days
28	20	$ 24,155	$ 27,054	$ 8,760	$ 9,811	9'x4'x4'	4.2	90 Days
42	30	$ 36,246	$ 40,595	$ 13,140	$ 14,717	12'x4'x4'	5.4	90 Days
56	40	$ 46,705	$ 52,310	$ 17,520	$ 19,622	14'x5'x5'	7.2	120 Days
70	50	$ 58,395	$ 65,402	$ 21,900	$ 24,529	16'x5'x5'	10.0	120 Days
84	60	$ 70,057	$ 78,463	$ 26,280	$ 29,434	16'x5'x5'	12.0	120 Days
112	80	$ 90,201	$101,035	$ 35,040	$ 39,245	16'x5'x5'	14.0	120 Days
140	100	$112,724	$126,250	$ 43,800	$ 49,055	20'x6'x6'	16.0	180 Days
280	200	$225,449	$252,502	$ 87,600	$ 98,112	20'x8'x8'	25.0	180 Days
420	300	$338,173	$378,753	$131,400	$147,169	22'x8'x8'	29.5	180 Days
560	400	$414,090	$463,780	$175,200	$196,225	24'x8'x8'	38.0	180 Days

Systems over 800 KVA are assembled on site at approximately $ 900 per KVA (over 205°F)
$1000 per KVA (under 205°F)

OPTIONS: Solid State Auto Speed Control----------------------------------$1,175.00
Salt Water Resistant Heat Exchangers----------------------------Engine Price Plus 25%
Stainless Steel Heat Exchangers---------------------------------Consult Factory
Non-Standard Instrumentation or Controls------------------------Consult Factory
Export Crating--No Charge

All prices are F.O.B. Factory. Terms (domestic) 1/3 deposit, 1/3 on shipment (inspection invited), Balance net 30 days. Terms (export) 50% deposit, balance on presentation of shipping documents. Direct payout figures supplied by Florida Power & Light Company, are based on cost to run equivalent HP electric motors at 5¢ per KWH.

TECHNICAL BULLETIN NO. 252W

ORMAT ENERGY CONVERTERS

FOR

ELECTRICAL POWER GENERATION

FROM

UNUSED INDUSTRIAL PROCESS HEAT

OR

GEOTHERMAL ENERGY

WHITING CORPORATION
15700 LATHROP AVENUE
HARVEY, ILL. 60426
TELEX: 2-53274

REVISION 7

FEBRUARY 1983

PREFACE

SYNOPSIS

Despite the rapid escalation of energy costs during the past decade, and a concerted, though somewhat belated, effort towards industrial energy conservation, very few systems producing electricity from waste heat have been installed due to the lack of a proven prime mover. Such a prime mover, the ORMAT Energy Converter (OEC), which is basically an Organic Rankine Cycle Turbogenerator (ORC), has, however, been widely used for fossil fuel power generation at remote sites because of its reliability. Now that waste heat recovery techniques have matured, it is being applied in this field. Technical risks are low, with key system components well developed and commercially available. The key to increased usage of ORC systems for industrial energy conservation lies in combining cost effective ORC system designs with well-selected applications.

The ORMAT Energy Converter makes it possible for industry to generate electrical power from low temperature heat sources not otherwise usable within the process plant. With the unique equipment package offered by ORMAT, the world's leading supplier of Organic Rankine Cycle (ORC) Turbogenerators, industry now has an alternative to the high and ever-increasing cost of electrical power. Virtually all process plants, including refineries, chemical, glass, primary metals, cement, pulp and paper, and food, reject vast quantities of previously unused low grade heat which may become a cost effective source of energy for electrical power.

In addition, the ORMAT Energy Converter can be used to generate electrical power from low temperature geothermal energy. It is now possible to utilize the previously wasted resource and use it to generate income. The system design enables the energy converter to be installed in isolated and remote areas.

As a fully-integrated and factory-tested system, installation of the ORMAT Energy Converter basically requires only connections to the heat source, the cooling water, the plant air supply, and the electric grid. System design eliminates any possibility of interference with the existing primary process.

1. SYSTEM OPERATION

The Ormat Energy Converter (OEC) is a self-contained, fully automatic equipment package, which generates electrical power utilizing otherwise unused process heat as a source of energy.

Like the steam turbine, the system is based on the Rankine power cycle but uses an organic working fluid which has the advantage of higher efficiency than steam at low heat source temperatures.

The working fluid, a fluorocarbon chosen according to heat source, is vaporized by the heat stream in the vaporizer. The pressurized vapor expands through the organic vapor turbine which is coupled to the generator. The generator produces electrical power with a 3-phase output voltage and frequency compatible with the plant power supply and the utility grid. The low pressure turbine exhaust is condensed in a condenser and the resultant fluid is pumped to the vaporizer by the working fluid cycle pump (see Figure 1).

The system is normally operated in parallel with the process stream, is fully regulated and incorporates a control system designed to preclude interference with the integrity of the basic process.

2. GENERAL DESCRIPTION

The OEC systems are supplied as skid mounted packages requiring minimum installation work on site.

2.1 Mechanical

The mechanical subsystem as supplied includes:

Vaporizer, preheater, condenser, receiver, turbine, totally enclosed generator, fluid cycle pump, organic fluid, automatic control valves, manually operated valves, check valves, safety valves, level switches, pressure gauges and pressure controls, complete internal piping connections which include working fluid cycle piping, pneumatic piping and lubrication subsystem piping.

2.2 Electrical

The electrical system is comprised of a free standing control panel which consists of a control section and a power section suitable for installation inside a remote plant control room, as detailed in section 3.6 herein.

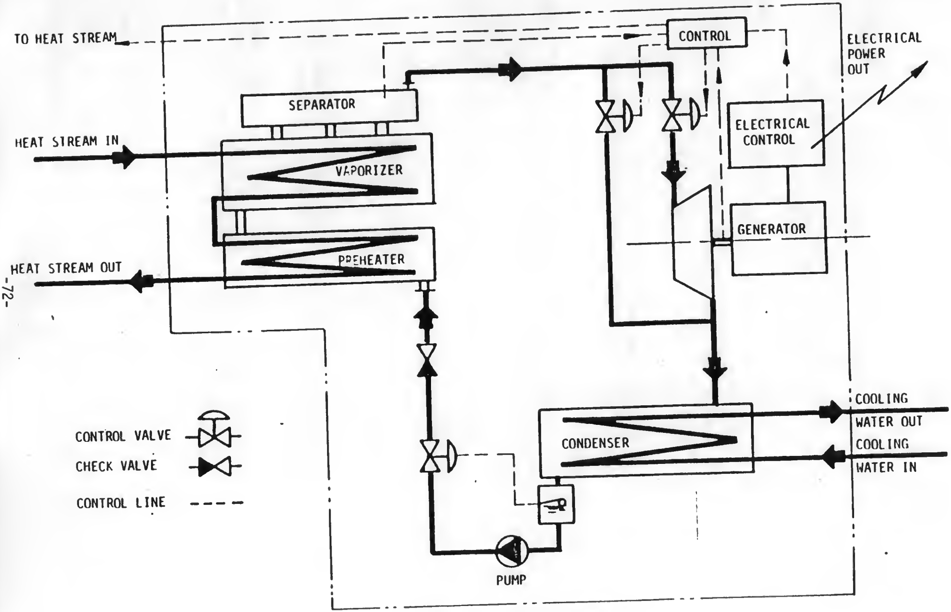


FIGURE 1 - SYSTEM SCHEMATIC
ORC FOR GENERATING ELECTRICAL POWER FROM PROCESS HEAT

3. COMPONENT DESCRIPTION

3.1 Turbine

The turbine is a single stage impulse turbine specially designed to work with organic fluid vapors at supersonic flow rates.

Each turbine undergoes an extended continuous operation test, performed at partial and full load, under simulated design conditions before being assembled into the system.

The turbine which is precision machined and balanced, utilizes lubricated bearings and double shaft seals.

The turbine lubrication system includes an oil tank, filters, pumps and valves. Oil is supplied to the mechanical seals and bearings with an adjusted flow rate.

The turbine control system will automatically shut down the operation when abnormal conditions occur, in accordance with the system safety logic design typically as shown in Figure 2.

3.2 Condenser

The condenser is a water cooled shell and tube heat exchanger designed and fabricated according to applicable standards and codes such as TEMA "C" and ASME codes for unfired pressure vessels.

A receiver tank beneath the condenser is provided, with level controls and switches to control the working fluid cycle pump operation.

The condenser is fabricated of carbon steel with tubes of Admiralty brass or other materials, as appropriate.

3.3 Preheater and Vaporizer

The preheater and vaporizer are shell and tube heat exchangers designed and fabricated according to applicable standards and codes such as TEMA "C" and ASME code for unfired pressure vessels.

Evaporation of the working fluorocarbon fluid takes place in the vaporizer shell side while the heating media flows inside the tubes. A liquid separator installed at the vaporizer outlet is designed to prevent drops of liquid from being entrained with the vapor and subsequently from impinging on the turbine impeller blades.

The preheater which preheats the organic working fluid before the fluid enters the vaporizer improves the overall efficiency of the thermodynamic cycle.

The vaporizer and preheater are fabricated of carbon steel. For corrosive heat sources appropriate alloys are used.

3.4 WORKING FLUID CYCLE PUMP

A motor driven multistage centrifugal pump is used for recycling the working fluid.

The pump is controlled by level controls placed in the receiver.

3.5 GENERATOR

3 Phase, brushless, induction type generator, close-coupled to the turbine, suitable for operation in Class 1, Div. 2 hazard areas.

Frequency:	50 or 60 Hz
Rated Voltage:	400 or 480 Volt, 3-phase, 3 wire, Delta connected, or other in accordance with clients requirements.
Protection:	TEFC, IP44, with air to air heat exchanger
Design:	NEMA frame, type C face mounted foot machine, IEC type B3/B14
Insulation:	Class F, 90°C temp. rise, according to NEMA MG1-1978 or IEC standards
Bearings:	Heavy duty grease lubricated ball bearings
High temperature protection:	3 PTC (thermistors) for the windings and 2 PTC for the bearings
Voltage regulating:	As the generator is of induction type, voltage regulation is inherent. After being linked with the mains, the generator maintains the same voltage as the mains
Compensating:	Power capacitors are connected to the generator to compensate for the reactive power needed for the excitation
Paralleling:	As the generator is of the induction type, paralleling is straight forward. The generator is brought to the no-load speed by the turbine under the synchronizer control and then linked with the mains.

NOTE: For applications where the OEC electrical system may not be linked with the grid mains a specially designed system may be supplied.

3.6 Control Panel

The control panel is a sheet metal NEMA type 4 enclosure for indoor and outdoor installation (not intended for operation in harzardous locations).

Operation is fully automatic and starting the unit is accomplished by merely depressing the "START" pushbutton.

The power section contains all circuit breakers, magnetic contactors, fuses, transformers, power capacitors, metering instruments, system protecting devices against overload, short-circuit and reverse power, mains and generator terminals.

The control section contains all system protecting and alarm devices, start and stop sequencing elements, turbogenerator accelerating and switching devices, indicating lights for failures and alarms, pushbuttons and switches for system operation, automatic battery charger and a synoptic diagram for operational sequencing. (See Figures 2 and 3).

The controls are powered by a 24 volt, 45 Ah battery (2 batteries 12 volt each) installed in a rack inside the panel.

The unit is automatically accelerated to the synchronous speed and then linked with the mains. An inrush current of up to 6 to 10 times rated current is drawn for a very short time (1 to 3 cycles at the power frequency) from the mains. The magnitude of this inrush current depends on the distance between the generator and the busbar and on the size of the power transformer feeding the busbar.

After being linked with the mains, the unit is in its normal operational mode and will automatically generate power.

In case of any failure, the system is automatically disconnected from the mains and either shut down or transfered to a mode of operation in which the system simply removes heat from the process stream. An alarm horn and indicating lights identify the mode of failure occured (see Figure 2).

Normal shut down of the system is executed by pushing the "STOP" pushbutton.

Cable entrance for mains and generator connections is from the bottom.

3.7 Hazardous Area Operation

For operation in hazardous areas, as defined by the National Electrical Code Class 1, Division 2, an optional equipment package is provided which includes explosion proofing of all skid mounted electrical components.

TYPICAL SYSTEMS SAFETY

ALARMS AND AUTOMATIC CONTROLS

PARAMETER	MALFUNCTION ALARM	TURBINE SHUT DOWN	TURBINE BY PASS OPEN 1	HEAT STREAM INLET CLOSED 2
GENERATOR-HIGH CURRENT	X	X	X	
GENERATOR-REVERSE POWER	X	X	X	
TURBINE GENERATOR-HIGH TEMP. BEARING	X	X	X	
VAPORIZER-HIGH PRESS.	X	X		X
VAPORIZER-LOW LEVEL	X	X		X
CONDENSER-HIGH PRESS.	X	X		X
CONDENSER-HIGH LEVEL	X	X		X
CONDENSER WATER FLOW TOO LOW	X	X		X
TURBINE-OVER SPEED	X	X	X	
LOW AIR PRESSURE	X			
TURBINE-LUB OIL HIGH LOW PRESSURE	X	X		X

NOTES:
1. ORC operating in heat removal mode only.
2. Process restored to original configuration.

FIG. 2

SYSTEM SAFETY LOGIC

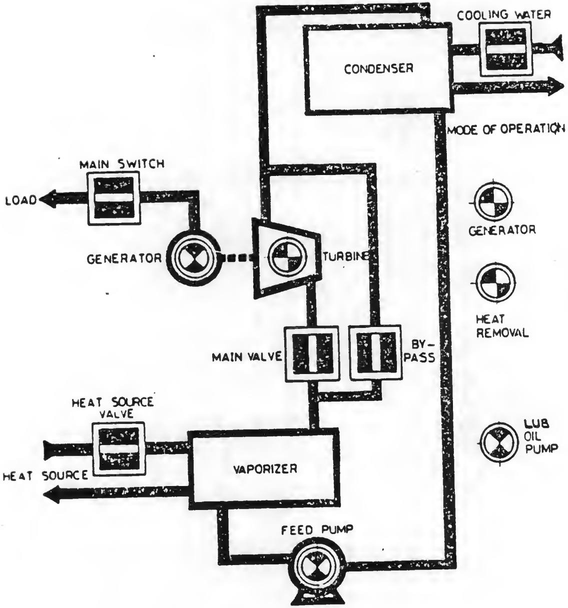


FIGURE 3

SYNOPTIC OPERATION CHART

4. Maintenance

In order to assure continuous operation of the ORC a periodic maintenance schedule is recommended, as follows:

4.1 Weekly Maintenance

a) Pressure, temperature, and electrical parameters.

b) Oil and working fluid level.

4.2 Monthly Maintenance (750 hrs.)

Visual leak inspection

4.3 Bi-Monthly Maintenance (1500 hrs.)

Generator lubrication.

4.4 Four-Month Maintenance(2500 hrs.)

Grease feed pump.

4.5 Yearly Maintenance (8000 hrs.)

Once a year the unit should be stopped in order to carry out a more detailed procedure which includes among other things:

- oil change
- filter replacement
- mechanical coupling checkout
- motive fluid leak-checking by means of leak detector

Heat exchanger cleaning should be carried out according to process and heating/cooling media, based on customer's experience.

4.6 Working Fluid

Normal operation should not cause fluid leakage, and the estimated quantity of organic working fluid that should be added, including liquid lost during this replacement of the fluid drying element, is 50 kg.

5. Application Criteria

For cost effective application, the heat transfer process should be in usable parameter ranges. The most important parameters to be considered are: source temperature, flow rate, viscosity, specific heat, latent heat and specific gravity.

Type of heat stream	Minimum temperature
Process liquids/hot water	93°C (200°F)
Condensing organic vapors	Most of the condensation should be above 120°C (250°F)
Steam	Temperature 100°C (212°F); Pressure: atmospheric

For specific applications an information form as in Figure 4 should be completed by the user and returned to our Applications Engineering Department.

6. Economic Criteria

Each user has his economic criteria which will be based on:

a) Local cost of electricity

b) Local criteria for return on investment

c) Interest rates

d) Government incentives for energy conservation

Other savings which can lower the installed cost per kW for the ORC equipment include:

a) Investment tax credits

b) Energy incentives

c) Saving in power from existing aircoolers which may not be required

d) Reduction in peak power demand charges

APPLICATION INFORMATION FOR RECOVERY OF UNUSED INDUSTRIAL PROCESS HEAT

Your Name: ____________________ **Title:** ________
Company: ____________________
Address: ____________________
____
Telephone No.: ________ **Telex No.:** ________

CHECK UNITS: British ☐ Metric ☐

1. HEAT STREAM DATA:

Composition ________
Check Type: Liquid .... ☐
Condensing Vapor .... ☐ •
Gas .... ☐
Flow Rate ________ kg/hr (lb/hr)
Heat Quantity ________ kCal/hr (BTU/hr)
Inlet Temp ________ °C (°F)
Discharge Temp ____ °C (°F)
Inlet Pressure ____ Bar (Psia)
Specific Heat ______ $\frac{kCal}{kg\text{-}^{\circ}C}$ $\left(\frac{BTU}{lb\text{-}^{\circ}F}\right)$
Latent Heat ________ kCal/kg (BTU/lb)
Viscosity (1) _____ CP at _____ °C (°F)
Viscosity (2) _____ CP at _____ °C (°F)
Specific gravity ____ at _____ °C (°F)

- • Please attach Condensing Curve if available

2. EXISTING HEAT EXCHANGERS: •

Type ____________
Check Material: Carbon Steel ☐
Stainless Steel ☐
Other Material: ________
Available Press. Drop _____ Bar (Psi)

- • Please attach Specification sheet if available

3. ELECTRICAL POWER DATA:

Plant Peak Demand - ______ kW
Energy Cost (Per kW-hr) Peak
______ Average
Voltage Required ________ V

4. COOLING WATER:

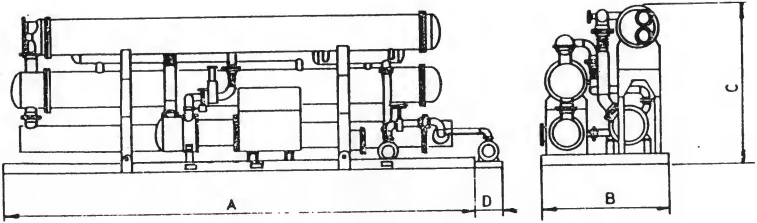
Source : Cooling Tower ☐ Other ____
Summer Temp ________ °C (°F)
Winter Temp ________ °C (°F)
Flow Rate Available ____ m^3/hr (GPM)

5. INSTALLATION FACTORS

Number of Similar Heat Sources in Company: ____________________
Availability of Heat Stream: ____________________ hrs/yr
Installation Considerations: ____________________
Unusual Factors: ____________________

APPENDIX 1

DATA SHEET AND TYPICAL APPLICATION SPECIFICATIONS



MODEL	NOMINAL CAPACITY	A	B	C	D	DRY WEIGHT
9000-300	300 kW	20'	8'	8'6"	2' 1½"	42,900 lbs
4000-300	300 kW	40'	8'	8'6"	-	67,000 lbs
1000-300	300 kW	40'	8'	8'6"	-	71,500 lbs
8000-600	600 kW	40'	8'	8'6"	-	92,500 lbs

Note: Actual output depends on the heat stream characteristics

ORC POWER SYSTEM DATA SHEET

TYPICAL APPLICATION SPECIFICATIONS

Heat Stream	Temperatures Inlet °F	Temperatures Inlet °C	Temperatures Outlet °F	Temperatures Outlet °C	Flow Rate 1000 lb/hr	Flow Rate 1000 kg/hr	Gross Output kW
Pressurized hot water	280	138	248	120	1,500	680	1,200
Heavy Distilate (22 API)	316	158	250	121	990	450	1,000
Light Gas oil (35 API)	315	137	227	108	242	110	338
Diesel Oil	306	152	235	113	1,100	500	1,260
Naphta	300	149	215	101	270	123	320
Kerosene	380	193	200	93	450	205	1,300
Condensing Steam	212	100	212	100	85	39	1,600
Condensing Hydrocarbon Vapors	315	175	225	107	96	44	620

STIRLING HISTORY

It is believed that 19th century inventors were motivated to build air engines because of a wide spread fear of steam boiler explosions. Early statistics are lacking, but from 1862 to 1879 there were over 10,000 boiler explosions in England, while in this country from 1880 to 1919 there were 14,281 recorded boiler explosions, resulting in over 10,000 deaths and 17,000 injuries.

By the term, "air engine" is meant a reciprocating heat engine which compresses cool air and expands heated air. The internal combustion engine is excluded because historic air engines did not employ combustion directly in the cylinder. Gases other than air were sometimes used, but the important distinction is that the gases did not change phase, as in steam powerplants. Air engines, generally, have been divided into three types: Type I were open cycle machines in which the products of combustion from a closed coal-burning chamber mixed with the working air and were admitted to the power cylinder through a valve. They are usually referred to as furnace gas engines, and approximated the Brayton cycle. Type II were closed cycle, externally heated machines, and are called Stirling or constant volume engines. Type III were open cycle, externally heated machines, and generally known as Ericsson or constant pressure engines.

It is difficult to review the history of the air engine without including the history of thermodynamics. The failure of the air engine to achieve the success of the steam engine during the 19th century can be ascribed to a misunderstanding of the nature of heat. Although Carnot wrote his "Reflections on the Motive Power of Heat" in 1824, acceptance of the kinetic theory of heat was not general until after 1860. The first experimental measurements of the mechanical equivalent of heat were published by Joule in England in 1843. Until this time, the caloric theory of heat was prevalent. Heat or caloric was considered a substance -- heat absorbed in the expansion of a gas became latent, but remained in the gas and was again evolved on compressing the gas. Tha caloric theory gave no explanation of the source of work produced by gas expanding against a piston. These errors resulted in many attempts to build perpetual motion heat engines during the 19th century. While the caloric theory also held back progress of steam power, it was easier for inventors to rationalize a steam-boiler-engine system than the more elusive air engine.

Robert Stirling was born in Pertshire, Scotland, in 1790. He was licensed by the Presbytery of Dumbarton in 1815, and was ordained September 19, 1816. Less than two months later, he applied for a patent for a regenerator and an air engine which included the regenerator. The title was, "Improvements for Diminishing the Consumption of Fuel, and in Particular an Engine Capable of Being Applied to the Moving of Machinery on a Principle Entirely New".

Stirling's grandfather, Michael Stirling, invented the first rotary threshing machine in 1756. Stirling's brother, James, was amember of the Institution of Civil Engineers and cooperated in the construction of air engines during the years from 1818 to 1847. Four sons of Robert Stirling attained distinction in the profession of engineering. The obituary notice of James stated that they were "descended from a highly respectable farming family, nearly every member of which, females included, possessed mechanical talent almost amounting to genius".

Robert Stirling became minister of Galston Parish Church, in southwest Scotland, in 1824, where he remained until his death in 1878. In 1842 the University of St. Andrews conferred upon him the honorary degree of Doctor of Divinity. Apparently he was esteemed as a minister, a classical scholar, as well as an inventor. References point out the difficulty of concluding which activity Stirling considered his vocation -- the ministry or engineering. Apparently he picked both, for about 25 years, by setting up a blacksmith shop adjacent to the manse. There he and James experimented with air engines, and as was reported, "also constructed with great neatness and dexterity many optical and other scientific instruments".

Although an explanation of the Stirling cycle has been given earlier, it is of interest to note several aspects of the original invention which were unique compared to other engines. The Stirling engine required no valves nor ports controlled by piston position. The power piston was located in the cool zone, thus assuring that it remained at a low temperature throughout the cyclic process from idle to full load. Operating on a closed cycle permitted the minimum cyclic pressure to be raised above atmospheric, which increased the specific output proportionally. This feature was credited to James Stirling, and the reslut was the brothers' most successful engine, which they installed at a Dundee foundry in 1843. The mean effective pressure and thermal efficiency of this engine exceeded that of any subsequent air engine for the next 95 years.

In their patent of 1827 the brothers anticipated the advantages of using a working fluid other than air and apparently had tested other gases. They stated "and though we generally work the engine with common atmospheric, yet whenever a supply can be conveniently obtained of nitrogen, carbonic acid (CO_2) or any permanent gas which does not readily corrode iron nor cause explosion, we use it with equal and in some respects greater advantage."

Many commercial air engines produced during the latter half of the 19th century eliminated the regenerator, or "economizer" as Robert Stirling preferred to call it. Apparently, this was done through misunderstanding its function or to simplify the design, but it sacraficed the high potential efficiency of the cycle. Although recognising the vital contribution of the regenerator, the Stirling brothers believed the exterior heating surface was only required to make up unavoidable thermal and friction losses. Robert stated in a paper in 1852, "the greatest enemy of the economizing principle, is the continual passage of caloric from the hot to cold parts of the engine, by radiation, conduction, and etc., which requires a continual supply of caloric to maintain the proper temperature of each." In a paper presented before the Institution of Civil Engineers in 1845, James also expressed the belief that the engine could have been a "perfect" engine except for friction and heat losses.

Because of these erroneous views, the heating surfaces of their engines were designed to include only the area of the hemispheric head of the displacer cylinder, with no provision for fins or projections to increase heat transfer area. With insufficient area for the power level, the safe working temperature of the cast iron heads was exceeded and they usually failed within less than a year of operation. The son, Patrick Stirling, reported to the Institution of Engineers of Scotland in 1860 that the heaters were, "kept red hot". After four such heaters had cracked, the Dundee foundry gave up the use of Stirling's last air engine and replaced it with a steam engine. Materials were available at that time which might have enabled the Stirling brothers to increase the heater area by means of tubes, as they had already successfully done with their rather sophisticated cooler designs.

Improvements on the original engine were patented jointly by the brothers in 1827 and again in 1840. Theri final engine ran the lathes and other machinery in the Dundee foundry from March, 1843 to January, 1847. The power cylinder inside diameter was 16 inches and piston stroke was four feet. Operating speed was 28 rpm. There were two displacer cylinders connected to the double-acting power piston. Each of the four foot diameter displacer cylinders contained an annular regenerator composed of thousands of 1/50th inch thick iron sheets, each measuring 38 by 1.75 inches, with a heat transfer area said to equal 3200 square feet. The minimum cyclic pressure at full load was about 150 psi and the maximum pressure was about 240 psi. A water cooled tube type heat exchanger appears to be located above the regenerator, much as in modern Stirling engine designs.

By means of a prony brake an average of 21 brake horse power was measured. The engine was said to consume 50 pounds of coal an hour. The specific consumption of 2.4 pounds per BHP hour compared to about 4 pounds for the best steam engines of the day. According to Stirling, the coal was of "inferior quality"; but if one assumes a heating value of 12,000 BTU/pound, the brake thermal efficiency is calculated to be 8.9 percent. If one then assumes a reasonable mechanical efficiency of 75%, and from additional data furnished by James Stirling on the heat to the cooling water, the indicated thermal efficiency appears to have been about 21% and the furnace efficiency about 57%. The temperature limits of the internal air were said to be 150°F and 650°F, but no measurements were known to have been made, except of the cooler metal temperature. The Carnot efficiency for this temperature range is 45%. Steam locomotives of the 1930's used from 3.5 to 6 pounds of coal per BHP hour.

MARTINI ENGINEERING

2303 Harris, Richland, Washington 99352
Telephone (509) 375-0115

25 August 1983

Mr. Robert Boucher
721 Desta
Hamilton, MT 59840

Dear Mr. Boucher,

I'm glad that Sunpower has decided to help you in your comparison of an organic Rankine engine generator and a Stirling engine generator for a wood-powered small electric power source. As you know, they have built a rice husk-powered ~~electric~~ power source to produce mechanical power for rice hulling in Asia. This they did on a development contract and they are still a number of years away from the time when such as engine can be purchased at a reasonable price and applied anywhere you wish. Both Stirling Power Systems at Ann Arbor, Michigan, and United Stirling Incorporated in Alexandria, Virginia, both of which are subsidiaries of United Stirling AB in Sweden, are on the verge of commercializating 20 kW and 40 kW Stirling engines. However, the first ones will be very expensive and it is not known at this time when engines can be had by anybody who wants one at a reasonable price. The engines that are now being built are built one at a time as needed and are used as prototypes for promising private development programs which have so far not been published. From my understanding of your telephone call, you are on a very modest budget and would not be able to afford one of the prototype engines. These engines also require heater temperatures in the 1200-1400 F range. A Stirling engine receives all its heat at this high temperature. There is no provision for receiving heat at any lower temperature. Therefore, the heat content of the gas leaving the heater needs to be salvaged by running it through an air preheater to preheat combustion air.

There have been tests done at United Stirling of burning sawdust in a cyclone combustor in which the hot gases from the combustor pass through an engine and then through an air preheater. Also, there have been some tests done at Stirling Power Systems and their parent company in Sweden in which straw was burned to produce hot gases which then ran through a heat exchanger. The air running countercurrent to these hot combustion gases then was used to heat the engine and what was left over was used as combustion air for the straw burner. In this way the gases passing over the engine are free of the particulate matter from the combustion. Therefore, there must be a problem of ash from the solid fuel combustion ending up on the heater tubes. The heat exchanger to heat the air from the engine can be well overdesigned to allow for the fowling of the heat exchanger surface by the ash.

Enclosed with this letter are two copies of the Stirling Engine Newsletter which report on the developments by the Swedes in using solid fuels in connection with the Stirling engines.

Stirling engines are eventually going to be better than Rankine cycle engines. However, they are commercially unavailable at this time and the development and use of this engine in connection with solid fuel should be held in reserve until development of the solid fuel combustor and the first generation Rankine cycle engine is well understood and developed and into commercial production. Then in the second generation one should consider using a Stirling engine. The combustion problems of clean burning solid fuel combustors would be well worked out by then and the Stirling machine would have time to come to market with a factory produced engine which can then be purchased and utilized for your purpose at a reasonable cost and with no strings attached.

I appreciate receiving your test reports on your burner. I generally understand how it works and I wish you success in your development. Sunpower is one of the best companies you could deal with because they believe in this type of application for the Stirling machine and have, as I have already mentioned, their own experience in solid fuel combustion. I would appreciate being kept informed as to your progress even though you might decide that Stirling engines are not for you at this time. It would be instructive to let the Stirling engine community know what went into this decision if you would care to report on it.

It seems from talking to you that you are continually making modifications in your burner design to improve efficiency and decrease costs and decrease emissions. I believe it would be folly to multiply your problems by incorporating an experimental engine with your experimental burner. It would be much more prudent at first to incorporate a more or less off the shelf engine even though it is not as efficient as it might be just so the concept can be demonstrated. Later on when this concept is working well, one can consider trying to find a more efficient engine which very well at that time could be a Stirling machine.

Sincerely,

W.R. Martini

WRM/df
Enclosures
SENL. NOV. 1981.
SENL MAY 1980.

DESIGN OF A LOW PRESSURE AIR ENGINE FOR THIRD WORLD USE

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Intersociety Energy Conversion
Engineering Conference
Los Angeles, California
August, 1982

DESIGN OF A LOW PRESSURE AIR ENGINE FOR THIRD WORLD USE

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Lawrence B. Penswick

Sunpower, Inc.
Athens, Ohio

Abstract

This paper reports on the design of a modern low pressure Stirling engine to produce 4 kW of shaft power. The engine is to burn rice husks for fuel and to be capable of manufacture in Third World countries. The power level is set by the requirement that the engine be capable of powering the local rice hullers. The main advantage of the Stirling engine in this application is that it is fired by rice husks instead of expensive and difficult to obtain petroleum based fuels.

Introduction

In recent times large amounts of time, effort, and money have been spent on the analysis and development of Stirling engines. In light of this fact it is surprising that the only practical engines commercially produced were built about 100 years ago. These engines were large and heavy for the power produced but did run on solid fuels and were safe to operate. They used air at low pressures for the working fluid, the seal problem was therefore a minor one.

Sunpower has long promoted the concept of a modern low pressure air engine (1,2). Using modern non-exotic heat resistant materials, such as common stainless steel, and present day understanding of the Stirling cycle, great improvements can be made over the Stirling engines of the last century. In applications requiring solid fuel and not limited by the need for high specific powers, the low pressure Stirling is a relatively simple, safe and efficient engine.

Because of these views, Sunpower welcomed a project with the Asia Foundation funded through the U.S. Agency for International Development. The objective of the project is to design a rice husk fired engine that can be produced in Bangladesh and is capable of powering a #4 huller, the smallest of the steel-roll rice hullers in common use there.

In order to investigate the local manufacturing capabilities, materials availabilities, and also to determine the actual power required by a #4 huller, a Sunpower engineer spent 3 weeks in Bangladesh during the initial stages of the program.

Visits to numerous local machine shops were made as well as to the local materials markets in Dacca. Power measurements were made on a diesel powered huller set up for testing at the local project site. An operating rural mill was also visited and power measured during normal milling operations. The result of the power study was the conclusion that approximately 4 kW would be required to operate a #4 huller.

The manufacturing facilities are very limited. The most common machine tools in use are lathes, shapers, grinders and drill presses with most of the equipment fairly old and ranging from poor to very bad in condition. There are very few milling machines in use and shapers are used for most milling type operations. As the cost of labor is relatively low, many operations, though time consuming, are done by hand. Steel sheet, for example, is cut in many instances with a cold chisel and a hammer.

Sand casting of aluminum and iron is common, but the materials used are often scrounged from any available scrap source making it virtually impossible to specify any particular alloy content. Thus, one of the major problems in producing a design for construction locally is that many shops will attempt to copy a machine and may substitute materials, depending upon availability, without regard to the effect on the design.

This project is thus not as easy as it might first appear to be. The design must be as simple as is technically feasible and there are many modern manufacturing techniques and materials that must be avoided.

Basic Engine Parameters

The basic engine design parameters are those factors which comprise the Beale number, namely speed, pressure and displacement. These were selected to simplify the engine while keeping the physical size and thus material costs within reason. Since the heat exchangers are usually the most complicated components of a Stirling engine special attention was given to them.

To reduce the complexity of the heat exchangers the design parameters tend toward a low speed and a high pressure. High pressure however places a greater demand on materials and construction techniques. Therefore, both a low speed and a low pressure were desired, implying that the physical size of the engine would be somewhat large. The

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large size is however beneficial in increasing the external surface area for heat transfer. Also, because the pressure is low the walls can be quite thin.

Ultimately, with the aid of computer simulation, the cylinder bore and stroke were selected at 30 cm. and 10 cm. respectively, giving a displaced piston volume of approximately 7 liters. The design speed became 720 RPM (12 hz.) with a mean cycle pressure of 4.5 bar (gauge).

Engine Layout and Mechanism Selection

There are numerous possibilities in the area of engine arrangements, a comprehensive listing of these is given by Walker (3). The drive mechanism is also considered when selecting an engine arrangement because obviously not all mechanism types are compatible with every engine arrangement.

A basic specification selected for the mechanism was to use a dry crankcase. In a dry crankcase there is no oil bath and all bearings are either sealed or made of self-lubricating materials. This decision was made for safety reasons, because the hazard of explosion exists if oil were to somehow get to the hot end of the engine. The use of a dry crankcase requires that the drive mechanism be designed to introduce only small side loads on the piston and displacer since the load capacity of self lubricating materials is usually small.

Since the displacement of this engine is quite large, the use of multiple working spaces was considered. The double acting engine, which inherently has multiple working spaces, was believed to be too complex for this program because of the ducting required to connect the separate working spaces with the heat exchangers. The multicylinder single acting engine type was also rejected because it requires a large number of component parts, especially in the mechanism part of the engine.

The main advantages of this concept appear to be in system packaging and smaller (but more numerous) component parts. Also, the torque delivered from such an engine to the flywheel is more uniform thus allowing for the use of a smaller flywheel. The complexity involved in such an engine including the larger number of working fluid seals was not believed to be warranted within the goals of the project. The multiple working space concept was thus abandoned for this project.

A simpler alternative is the single working space type of engine. Here again several options exist as to engine layout. Three basic types of engines exist which have been identified by Walker (3) as two piston machines, piston-displacer in separate cylinders and single cylinder piston-displacer machines.

Two piston and piston-displacer in separate cylinder machines can be considered together when selecting the drive mechanism. The drives for these two types of engines can usually be less complicated than for a single cylinder arrangement. This arises partially from the fact that the axes of the two moving components are non-coincident which in turn reduces the need for duplicate and "forked" mechanism parts to insure symmetrical loadings on the piston. The least complicated drive for these types of engines is probably the Rose (4) type of mechanism.

These two cylinder arrangements do have some disadvantages. In the two piston machine both pistons and their seals are subjected to the cyclic pressure variations of the working space and in fact there is a large energy transfer through the mechanism from the expansion space to the compression space. This problem is alleviated in the two cylinder piston-displacer type of arrangement, but at the expense of a dead volume penalty since the compression space cannot be reduced to zero volume at any point in the cycle. Two cylinder arrangements are also generally more complicated from a structural point of view than is a single cylinder arrangement.

The single cylinder piston-displacer engine thus has several advantages compared to the other possibilities. The main drawback here is that most of the possible mechanisms have a greater number of separate parts. Some of these are however lightly loaded as in the displacer drive. Also, the duplicate parts used to provide symmetrical piston loads each carry only half the load as would a single drive link.

Many possible mechanisms were considered for the single cylinder type of engine. The least complicated of which is the Ringbom drive. In the Ringbom the piston is mechanically linked to the crankshaft while the displacer is driven by the cyclic gas forces in a manner similar to the Beale free piston engine. This type of drive is however still in the developmental stage.

Two of the least complicated single cylinder drive mechanisms are those devised and presented by Senft (5). These are the "tape drive" and the "inclined yoke" mechanisms. Of the two, the tape drive is probably the best suited for a dry crankcase.

The intent of the program is that the final design be available to any country for unrestricted manufacture, therefore, these two and other patented mechanisms could not be used

Many other mechanisms were considered, with the final selection being an Ericsson type of linkage. This linkage is the one used on the commercially produced Ericsson (Stirling cycle) engines of the 19th century. During the layout design of the mechanism, manipulations were made to the arrangement of the Ericsson in order to achieve a more compact design. Quite interestingly, the mechanism evolved to look much like the familiar Philips bellcrank drive except for the additional piston side link. This side link exists to reduce the side loads produced by the piston connecting rod. An assembly drawing of the engine and mechanism is given in Figure 1. With due respect for Ericsson, Sunpower has chosen to continue to refer to this mechanism as an Ericsson type of linkage.

Burner Design

The rice hull burner development for the AID engine represented a number of technical problems unique in the application of bio-waste products as a fuel. Not only was the combustor expected to conform to the basic requirement of local manufacture but also to operate at a reasonable efficiency and, most importantly, relatively high temperatures. The last factor eliminated from consideration the majority of current and past projects on direct combustion of bio-wastes, since in most cases the outlet temperatures have fallen in the range of 250 to 600° C.

The initial engine design indicated the following furnace characteristics would be required:

Power to engine heater	30 to 40 kW
Heater inlet gas temperature (no air pre-heating)	900 to 1000° C
Fuel consumption	15 to 20 kg/hr

After a review of available information (5) an air suspended cyclone type furnace was selected as the most probable type to meet these requirements. However, a number of critical parameters; combustion efficiency, burner stay times, air-fuel ratios, and basic furnace geometry remained unknown. Because of this it was decided early in the program to construct a test burner that would allow a wide variation of as many of these factors as possible. The test furnace was constructed of stacked masonary in the lower portion with a conical stainless steel upper portion in which an engine head simulator could be inserted.

Initial testing pointed out a number of problems ranging from long transient times before full power was reached, to that of feeding rice hulls at a uniform rate into the furnace. But recent testing has maintained temperatures of 1000° C and above at better than expected combustion efficiencies. From these tests, furnace geometry and size, ash separation techniques, and necessary cyclone geometry have been determined. This information has been employed in the final burner design shown in Figure 2.

This design will employ air preheating to reduce furnace losses and the addition of secondary air into both the air downstream of the ash separation section and into the ash tray itself. Fabrication will be of insulating fire brick laid up in steel forms which make up the structural frame of the furnace. Because of the excellent insulating characteristics of the ash itself, the entire assembly can be surrounded by a layer of ash to further reduce heat losses.

Heater Head Construction

The design of the heater head was governed primarily by the material available for its construction, (304 stainless steel) and the methods that could be used for manufacture. The construction method settled on was to roll and seam weld stainless sheet into a can, dome a cap by means of a clamping ring and a hydraulic pump, and weld the dome to a stiffening ring welded to the end of the can. External fins are made by tack welding stainless strips in place. Internal fins are crimped thin stainless steel sheet formed like an accordion and stuffed between an inner liner and the main heater can. Thermal contact on the heat exchangers is enhanced by furnace brazing filler between the heater can wall and the fins in an oxygen atmosphere. The size of the head is dictated mostly by the need to get 30 kW of heat into the engine at the combustion gas temperature (1000° C) obtained from the rice husk furnace.

Displacer and Piston Construction

The displacer body is a rolled and seam welded can with a formed dome welded to a stiffening ring on the can just like the heater head.

Thin stainless heat shields, tack welded to form a conical section are tacked on the inside. The can is glued with epoxy to a cast aluminum displacer cap. This is finished to a fairly close fit with the cylinder, and uses a self lubricating material for the seal.

The piston is an aluminum casting fitted with a leather cup seal.

Regenerator

The regenerator consists of .11 mm stainless wire packed to an 8 percent density. Preliminary engine shakedown and testing will be performed with a Metex regenerator. Subsequent testing will be made with simpler materials such as crimped wire and dimpled foil.

Cylinder Construction

The cylinder will consist of a cast aluminum cooler shell with a cast iron cylinder liner held in place by means of a shrink fit. The internal cooling passages will be formed using a shaper before the liner is fitted in place. Cooling will be provided by water flowing through passages shaped on the outside of the cylinder, with a removable water jacket for cleaning.

The cast aluminum cylinder is designed such that its strength will be adequate even if pure aluminum is used in the casting.

Crankcase Construction

The crankcase for the prototype is made from commercially available tube to which a hydraulically formed cap is welded. "Ears" have been added to the tube to provide pivot points for the mechanism without having to resort to a larger crankcase. To support the crankshaft, gussets are welded inside and out between the crankcase housing and a main crankshaft supporting tube. Of course, all final machining is performed after fabrication.

The size of the crankcase is such that all mechanism assembly including attachment of the piston and displacer are performed before bolting the crankcase to the engine. The absence of a crankcase cover prevents the possibility of running the engine in the dusty environment with an open crankcase and also contributes to the safety of the device.

Closure

The basic design and construction techniques of a hot air engine have been presented. The details of this design have been governed by the requirement that the engine be capable of manufacture in Third World countries. As of the writing of this paper, the final parts for the prototype are being machined. The specific details and performance of this engine are to be reported in a futurs paper.

Acknowledgement

Sunpower wishes to acknowledge the comments and aid given by the outside consultants on this program; Graham Walker on engine design, and Eldon Beagle on the rice hull burner design. L. Merrick Lockwood, the project manager for the Asia Foundation in Bangladesh provided the liason with local manufacturers and instruction in local technology.

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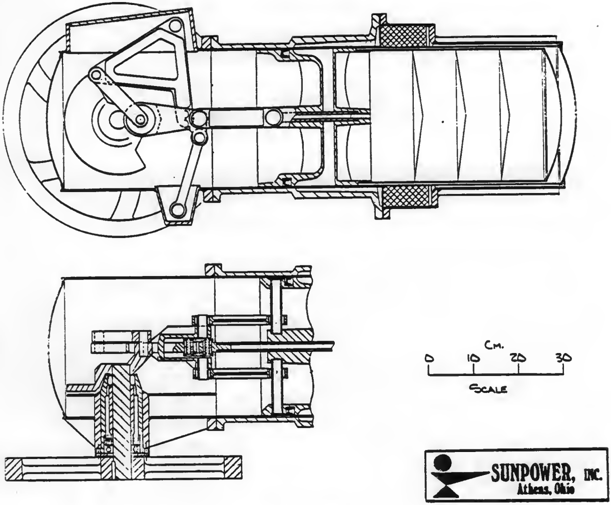

SUNPOWER, INC.
Athens, Ohio

FIGURE 1 ENGINE ASSEMBLY

1) PRIMARY COMBUSTION SECTION
2) ASH SEPARATION
3) SECONDARY AIR MANIFOLD
4) SECONDARY COMBUSTION SECTION
5) ENGINE HEATER HEAD
6) ASH REMOVAL

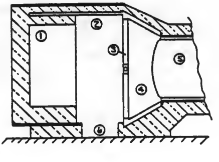

FIGURE 2 BURNER DESIGN

BIOMASS FUELED STIRLING ENGINE COMBUSTOR RESEARCH

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ABSTRACT

This paper is a progress report on a 3 phase research program to investigate combustion of wood chips in a Stirling engine. The program objective was to achieve efficient combustion without fouling the engine heat exchanger.

The Stirling cycle is described as well as 4 engines designed by United Stirling. Four Phase III tasks are listed. Only Task 1, a literature survey, and Task 2, combustor analysis and testing, have been completed.

Insufficient literature references resulted in modifying task 1 to study wood burning systems and fuel parameters, in general. These are discussed as well as the special characteristics of Stirling combustion systems. The rationale for selecting a 2 stage combustor to facilitate ash handling is explained.

Under task 2 both a cyclone and a stoker type gasifier were designed and tested, for stage one. The cyclone type was tested for cold particle separating efficiency, followed by free-burning tests. Concentration of CO was 7.5 to 15 percent. Air, fuel and ash controls are discussed. The stoker gasifier involving a rotating grate is described. Initial gasification testing and parameter selection have been made. The best design will be tested with stage 2 and integrated with a model 4-95 Stirling engine in May.

INTRODUCTION

This paper presents an overview of a research program at United Stirling AB (USAB) of Malmo, Sweden, to investigate combustion of wood chips in a Stirling engine. USAB is a subcontractor to United Stirling Inc. (USI) of Alexandria, Virginia. The present program is the 3rd phase of a DOE/Battelle NWL contract with USI which began in 1981. Phase I and II were reviewed at the 13th and 14th Biomass Contractors Meetings.

Phase I work verified the feasibility of metering and burning wood chips, up to 8mm length, directly in a cyclone combustor which simulated a Stirling engine burner; but the combustor was not tested in an engine.

Phase II dealt with indirect combustion in a continuous flow biomass (wood chips) gasifier connected to a USAB Stirling engine. Automatic control of fuel/air flow and engine heat exchanger temperature was accomplished. Net conversion efficiency was about 20% and maximum power achieved was 24 kW. Exhaust emissions were acceptable and could be further reduced by optimization of controls and components.

This report covers progress from September through January, of Phase III, a 13 month research program to analyze, design, fabricate and test an integrated Stirling engine solid biomass fuel combustion system. Its main objective is to achieve ash and particulate removal ahead of the Stirling engine heat exchangers, so as to prevent fouling.

BACKGROUND

United Stirling, Inc. (USI), a wholly owned subsidiary of USAB, was established in 1979 to facilitate Stirling engine technology transfer, with the ultimate objective of manufacturing Stirling engines in the United States. It was the recognition of the Stirling engine's many attractive features which led to the formation of United Stirling in 1968:

- high thermal efficiency
- low noise and vibration
- low exhaust emissions
- no lubricant consumption
- long engine life
- multi-heat capability

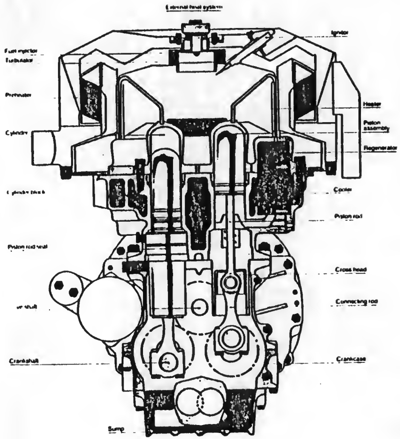
The Stirling engine is an external heat engine. It differs from the more familiar internal combustion engines by its ability to accept a wide range of heat sources at the proper temperature, including solar radiation and continuous, external combustion of solid, as well as, liquid and gaseous fuels. The engine operates on a closed, thermally regenerative cycle which results in an ideal efficiency equal to the Carnot. It is a valveless engine which uses pairs of pistons in the right phase relation to compress and expand a gas, usually helium or hydrogen, to produce power. Heat is rejected from the cycle to a cooling water circuit.

In 1980, an in-house program at United Stirling investigated the potential of wood fuel in a Stirling engine. Successful burning of wood particles was achieved in a cyclone type combustion chamber, first, in a free-burning test, and later when integrated with a model 4-95 engine developed by USAB. A cross-section of the 4-95 engine is seen in figure 1. Products of combustion passed around the tubes of the engine's standard involute heat exchanger, shown in figure 2. Fuel particle size included ground poplar powder, sawdust and conifer chips up to 4mm length. Satisfactory combustion and engine operation resulted with negligible carryover of particulates. However, evidence of clogging of the heat exchanger was noted, as well as coating of the cyclone combustor walls by melted ash.

The 4-95 Stirling engine is a 4 cylinder, double-acting design having a displacement of 95 cc per cylinder and a maximum power of approximately 40 kW at 4000 rpm. It was originally designed as a laboratory test engine. Following the 4-95 base engine lead, United Stirling has been developing two other 4 cylinder engines; the 4-275 which is capable of 85 kW at 2200 rpm and was designed as a heavy duty industrial engine; and the ASE Mod I which has been developed under the DOE/NASA Automotive Stirling Engine program, and has a maximum output of 54 kW at 4000 rpm. In addition, a smaller two cylinder "V" engine generator set has been developed, aimed at total energy applications. It is rated at 15 kW at 3600 rpm. All four engines have the potential for operating with solid biomass fuels.

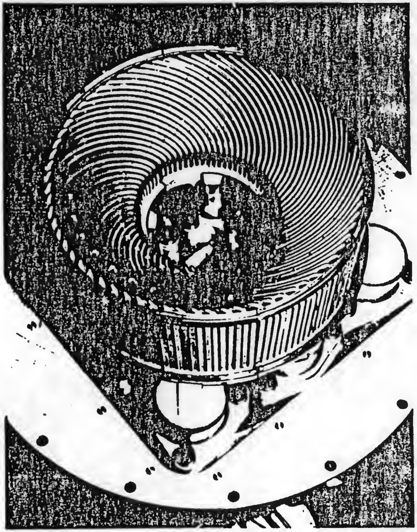
PHASE III - AN INTEGRATED BIOMASS COMBUSTOR

The scope of work under phase III called for United Stirling to conduct a research program to analyze, design, fabricate and test a prototype biomass combustor with continuous ash handling capabilities; follow-



Cross Section of the 4-95 Engine

Figure 1



Involute Heat Exchanger

Figure 2

ing this, to integrate the combustor with a Stirling engine, measure system performance, and maintenance and cleaning requirements of the ash handling system and the engine heat exchanger.

Four major tasks were included in the proposed effort:

Task 1. — Literature survey, emphasizing ash and slag removal methods.

Task 2. — Combustor concept analysis and testing. Several concepts were to be considered.

Task 3. — Design, fabrication and performance testing of the most promising concept from task 2, for a minimum of 6 hours.

Task 4. — Integration of the combustor system from task 3 with the 4-95 Stirling engine. The testing is to be made at an average load level not less than 20 kW (gross power), during a total test time of not less than 100 hours (not continuous). The time between major cleanings of the engine heat exchanger and/or combustor will be determined, as well as the best procedure for cleaning.

PHASE III - ACCOMPLISHMENTS

Task 1 - Literature Survey

The purpose was to survey the work reported in the technical literature concerning solid biomass combustion - emphasizing ash/slag removal methods and system. It was of special interest to note any ash-handling system in the power range of this program i.e. thermal power of approximately 100 kW; also if there were systems using preheated air in the order of 700°C.

The computerized searching was based on the Pascal and Compendex data bases, using combinations of key-words such as Biomass, Combustion, Solid fuel, Ash, Slag and Removal. In conclusion, the search was disappointing, resulting in very few usable references. Therefore, it was decided that the survey should be more general in its content and discuss different wood burning systems and the most important parameters for selecting a solid combustion system for the Stirling engine. Some of these parameters are: fuel size, moisture content, ash content and the slagging temperature.

Fuel Size

One important parameter in discussing solid biomass fuels is the fuel particle size. This is vital for selection of a fuel handling system.

Whereas agricultural crops may require little or no processing for conveying and burning, forest crops usually require a great deal of processing. The extreme in processing results in a powder, which might come from either crop residues or from trees. Generally, however, trees for commerical, non structural purposes are reduced to chips. Automation of the combustion system also requires certain processing of the biomass regardless of combustion needs, because the fuel handling devices require a fairly constant fuel size in a feedable form.

Moisture and Ash Content

Water content (wet basis) of fuels ranges up to approximately 60% for green wood. It has a strong influence on the heating value of the fuel. For example, the gross heating value of a typical wood fuel decreases from 7440 Btu/lb to 3500 Btu/lb as the moisture content rises from 15 to 60 percent. It also influences the flame temperature and consequently the capacity to transfer heat at a fixed heat exchanger temperature level.

The ash content ranges from less than 1% for most woods to over 5% for some agricultural crops. The ash content has a negligible influence on combustion, per se, but the ash must be removed in some manner, either before or after combustion occurs, to prevent blockage of the system. An important characteristic of the ash is the softening point or slagging temperature. This temperature is for many fuels in the order of 1000°C. At even higher temperatures the ash becomes liquid. In designing an ash handling system it is necessary to decide either to operate below the softening temperature (solid ash) or above the fluid temperature (liquid ash). In between these temperatures the ash is viscous and difficult to handle.

Biomass Combustion Systems — General

All biomass sources have the potential for either direct or indirect combustion — conversion to alcohols and pyrolytic oils and by gasification. Direct combustion systems for solid biomass fuels are almost always slightly modified coal combustion systems. These have included four major systems: 1) Fuel bed or stoker, 2) Suspension or pulverized, 3) Cyclone and 4) Fluidized bed.

Concentrating on systems in the power range of 100 kW one can exclude the fluidized bed for two reasons. First, the bed temperature is too low for heat transfer to the Stirling engine, and second, the heat exchanger in the Stirling engine is a convective type and not suitable for emersion in a fluid bed.

In the stoker system the relatively coarse fuel is resting on a bed surface and is burned off from the top. In the pulverized method the fine coal particles are in suspension with the air for combustion. The cyclone system uses coarser fuel and depends on centrifugal forces to separate the particle sizes during combustion. In all such combustion systems the fuel could be burned in one stage or in several stages.

Stirling Engine Biomass Combustion System

Energy is supplied to the Stirling engine by heat transfer from combustion gases to a heat exchanger which operates at a temperature level of about 750°C. It is a close spaced tubular, two-pass exchanger comprising a plain tube first-pass, and a finned tube second-pass. The combustion gases exiting from the second-pass exchanger are at a temperature level of approximately 800°C. These gases enter the air preheater and transfer heat to the air for combustion, which is raised to about 700°C. This results in a combustion temperature of over 2000°C, which is considerably higher than found in steam boiler or gas turbine combustion systems.

One of the conclusions from the abortive literature survey was that it would be extremely difficult if not impossible to attempt to burn wood chips directly in a kind of cyclone combustor. Because of the high combustion temperature the combustor would require ceramic walls and at least two additional all refractory cyclones in series to assure that the melted ash and particulates could be separated out ahead of the engine heat exchanger. It was also believed that the pressure drop through the cyclones would result in prohibitively high combustion blower power.

As a consequence, it was proposed to separate the biomass combustion process into two main stages for improved ash and particulate handling, to insure more complete combustion and to minimize emissions of oxides of nitrogen. The first stage involves fuel-rich combustion with ambient air inlet, and the generation of a combustible gas; the second stage involves preheated secondary air admission and a higher turbulence level to complete the combustion process and aid the removal of particulates. The rationale for this was: Use the lowest possible temperature in stage one to insure solid ash handling, followed by relatively clean gas burning close to the engine heat exchanger, and at the same time utilize the available preheated air so as to maintain the highest possible overall thermal efficiency.

The two stages were to be comprised of two insulated chambers, one above the other, with the model 4-95 Stirling engine mounted inverted above the second stage, as shown schematically in figure 3. The engine oil system was adapted froma 4-95 solar Stirling engine, already modified for inverted operation in a parabolic dish concentrator.

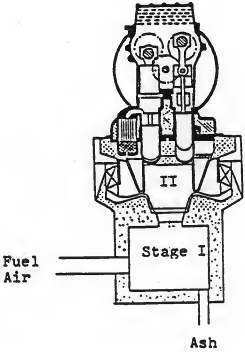


Figure 3 - Two Stage Biomass Combustor and Model 4-95 Stirling Engine

Task 2 - First Stage Analysis

One of the purposes of task 2 was to analyse the first stage flow pattern and separation capability.

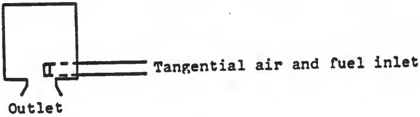
It was decided that two different concepts should be investigated.

A - cyclone type gasifier

B - stoker type gasifier

A - Cyclone Gasifier

The goal behind the cyclone gasifier was to produce gasification in a compact device, shown below schematically, which also acts as a separator.

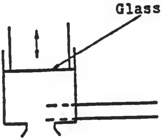


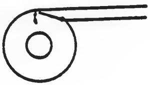
To act as a gasifier the air/fuel flow ratio should be decreased to a level well below stoichiometric.

As the cyclone itself is a separating device the coarser particles will stay in the cyclone, either circulating at the closed end or, if small enough, remaining on the wall surface. Unlike fixed bed gasifiers the cyclone type does not have any separate zones for drying, oxidation and reduction. Of course, these processes will occur, but at different locations due to the turbulent flow and velocity differences between particles and gases. The CO content for this type of gasifier was expected to average about 10%, rather than the more typical 20% range expected from gasifiers developed for IC-engines. However, this should be satisfactory for the Stirling engine as the gas will be burned with 700^{0}C preheated air.

Separation Efficiency

The design work started with tests aimed at selection of size parameters from the point of view of separation, under room temperature conditions. A test cyclone was built with a diameter of 250 mm, which included an adjustable length and air inlet area, as shown below. The flow patterns could be studied through the adjustable glass cover.





To simulate ash particles wood powder was used. Powder size ranged from about 50 microns to a maximum of 240 microns. For convenience, the cyclone outlet was placed vertically downward during these separation tests. The effect of gravity compared to drag forces is negligible.

Separation efficiency was defined as: That percent of total mass input which is collected on the cylindrical part of the combustor and on the lid. The tests show that most of the particles are trapped in the corner between the cylinder and the lid.

The tests were performed as a function of air flow, corresponding to 25, 50, 75, 100 and 200% load. Not knowing the proper air to fuel ratio when firing this gasifier, an air excess factor of 0.6 was assumed for these tests. The air inlet areas selected were 5, 10, 20, and 30 sq. cm. Combustor lengths tested were 100, 200 and 274mm. The diameter of the cyclone was chosen as a result of previous experience with cyclone combustors.

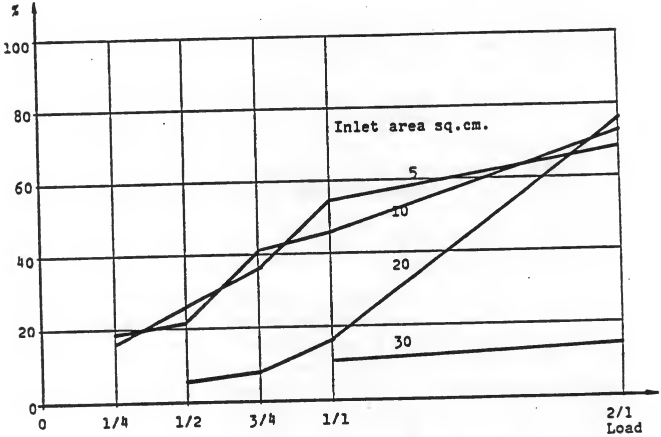
Results of Cold Testing

The results show that efficiency increases with increasing inlet air velocity (increasing load). See figure 4 which presents separation efficiency over the load range for the 200 mm length. However, the results are to be treated qualitatively as the condition in the gasifier is different when firing. Since the velocity will increase when heat is released, the separation efficiency should increase. Furthermore, the separation load (the amount of ash per unit time) will be much less during combustion as the ash content in the fuel to be used is only about 0.5%.

The conclusion from these tests was to use a combustor with an air inlet area of 10 sq.cm. and with a length of 200 mm. Pressure drop for various parameters is given in figure 5. Quantitive determination of the separation efficiency was to be done during free burning tests.

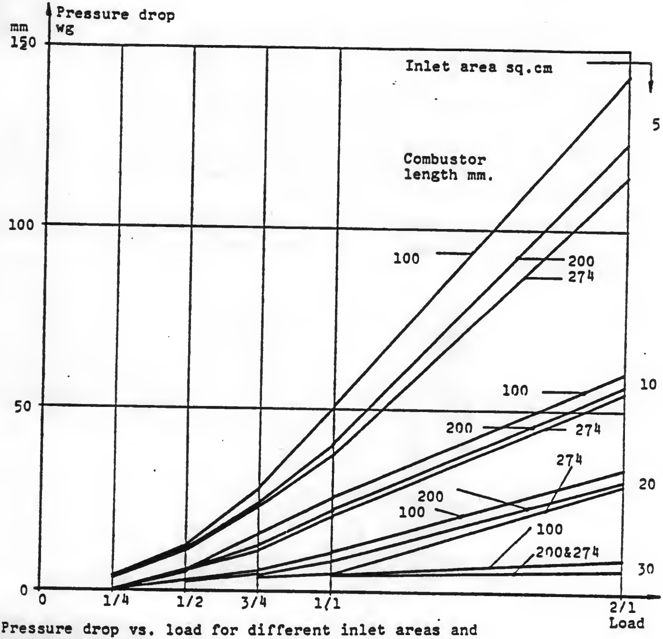
Free Burning Tests -- Preliminary Results

The cyclone gasifier test rig was then modified for free burning tests with wood chips as fuel. A sheet metal lid replaced the glass one, and a start up system using diesel fuel was incorporated. The free burning tests have been performed with the cyclone gasifier



Separation efficiency vs. load. Combustor length 200 mm.

Figure 4



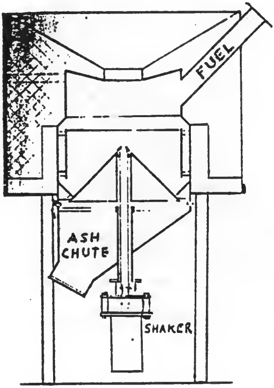
Pressure drop vs. load for different inlet areas and combustor length.

Figure 5

pointing upwards, and the wood chips were fed to the gasifier through the gas outlet. A combustor 200 mm long with different air inlet areas was tested at several loads.

The results show that carbon monoxide concentrations in the range 7.5-15% could be achieved. This should permit burning with 700°C preheated air in a second stage combustor. However, some of the longer fuel particles were not in suspension with the air but, especially at low loads, were resting on the lid. As the load increased more and more particles began to move and circle around in the gasifier. A higher velocity must be used to assure that the particles remain in suspension with the air.

An ash handling system for the cyclone gasifier is shown in the figure below. The ashes are discharged by the vibration of the conical bottom mechanism to the ash chute.

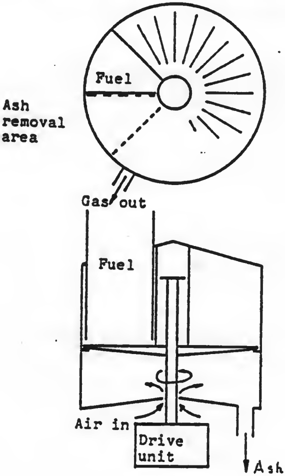


The fuel enters from the upper right corner. The air enters through four tangential pipes near the outlet end. Tests of the complete cyclone combustor with its second stage gas burner are expected to begin in April.

Control system design has been completed but all components have not been fabricated. The design calls for an electronic control box, several sensors and servo-motors. A temperature signal from the Stirling engine controls gasifier air flow; for example, a drop in temperature of the engine's heat exchanger results in higher air flow through the gasifier, and vice versa. Another thermocouple in the gasifier outlet region modulates the flow of wood chips into the gasifier.

B - Stoker Gasifier

The other first stage concept being studied is a stoker type of gasifier. The design is based on previous Phase I experience and test results. The concept is shown in the schematic drawing.



STOKER GASIFIER

Design and Function

The combustion chamber has a 460 mm diameter rotating grate consisting of 24 stainless steel bars which slide at their outer end on a projecting ring type cam. The fuel inlet is at one part of the circle. As the bard pass over an open area once per revolution, the outer end of each bar follows the ring cam profile and drops down approximately 3 inches. This sudden motion causes the ash particles to shake off. During a further rotation of 45 degrees, the grate bars gradually rise to their original level. Air for gasification enters through the hollow central drive shaft. In the ash pit a rotating scraper diverts the ashes to an outlet.

The rotating grate drive includes a variable speed D.C. drive motor, a gear box, a chain and sprokets combined with an overload clutch. A bed level sensor controls the motor switch.

Testing

Testing of the stoker gasifier has included start-up tests as well as operation for extended periods at normal operating temperature. An early problem encountered was galling and sticking of the grate bars on the lifting section of the ring cam. This was corrected by reducing the slope of the cam and increasing the torque level of the clutch release mechanism.

Initial gasification tests have been performed. Parameters investigated included height of the fuel chute outlet above the grate level versus fuel bed depth, and the effect of bed depth on the gas quality. Chute heights ranged from 3 to 15 cm, with 9 cm giving the best results. A bed depth of 6 cm resulted in a CO concentration of about 15%, a load capacity of 110 kW and a bed temperature of about 800°C. This appears to be nearly optimum, although much additional testing is required when the entire system has been assembled. The initial test bed set-up is shown in the photograph, figure 6.

It is expected that further modifications and testing of both the cyclone type and stoker type of gasifier will be made during March and April. One of the two systems will then be selected for integration with a model 4-95 Stirling engine. How each system would appear is shown schematically in figure 7.

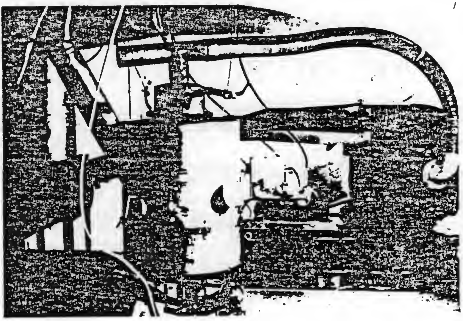
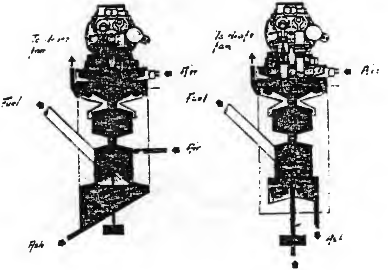


Figure 6 - Stoker Gasifier Test Hardware

Poor original



Cyclone

Stoker

Figure 7

THE DEVELOPMENT OF A 1 kW ELECTRICAL OUTPUT FREE PISTON STIRLING ENGINE ALTERNATOR UNIT

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THE DEVELOPMENT OF A 1 KW ELECTRICAL OUTPUT FREE PISTON STIRLING ENGINE ALTERNATOR UNIT

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Abstract

Sunpower, Inc. has been engaged in the development of free-piston Stirling engine alternator units since 1971. After considerable experience gained with high pressure (50 to 80 bar) engines, it appears that the high pressure configuration may not necessarily be the most appropriate approach to a successful commercial design. The repeated difficulty associated with the design of an efficient, cost effective heat input system plus the tight clearances forced by a critical sensitivity to leakage losses poses problems which have not been satisfactorily resolved in the past. In an effort to address these and other specific technical problems relating to the commercialization of Stirling engines, Sunpower has decided on a radical departure from the high pressure design so prevelant in the industry. Recent engines built at Sunpower all operate at substantially lower charge pressure than those built in the past. Described here is one program to develop a low pressure 1 kW free-piston alternator unit. It is shown that the low pressure configuration addresses both the heat input system and clearance problems with an acceptable compromise in the other aspects of the design. Two generations of the design have been built, one has been operated on a coal burner with very encouraging results and the other is presently on test in our laboratory.

Introduction

Development of Stirling cycle related technology has been an ongoing effort at Sunpower, Inc. since 1964. During this time there has always been a considerable interest in free-piston linear alternator applications. The first Stirling powered linear alternator was built in 1973. This was a simple machine of low output power (10W) which was later to become Sunpower's first commercial design, the Model 10. Following soon after was a 100 W design known as the M100 which was also sold commercially in limited quantities (1). Both the Model 10 and M100 are shown in Figure 1. In 1976 work began on a larger machine of 1 kW output. This design, the RE-1000, was specifically a research engine to evaluate the technology for higher power levels. The RE-1000 is a high pressure engine (70 bar) with an electrically heated head (1), (2), (3). Seven RE-1000 machines were built and delivered to customers interested in this technology. None of these RE-1000 machines were delivered with alternators, the power being dissipated in a mechanical dashpot load. A typical efficiency at the design point was better than 30% based on heat rejected. Figure 2 gives some idea of the size and geometry of the RE-1000.

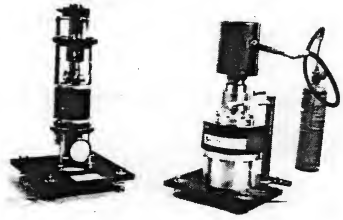

Figure 1 Model 10 and M100

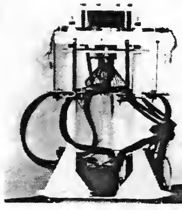

Figure 2 RE-1000

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duction in the appendix loss by at least 30%. It seems possible to reduce the loss even further, however, at 30% reduction the effect of appendix loss on engine efficiency is tolerable. Figure 7 shows the effect of insulating the appendix gap.

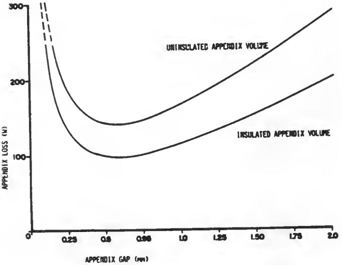

Figure 7 - Effect of Insulating The Appendix Volume

In the design effort, there has been an emphasis on keeping the overall assembly mass low. The final engine and alternator mass for SPIKE 1 is 25 kg and for SPIKE 2 it is closer to 27 kg. A negligible amount of plastic has been used. In a production machine plastic would be used wherever feasible resulting in a projected mass saving of 7 kg at least. Figure 8 shows SPIKE 1 and SPIKE 2 at the same scale.

Table 1 gives an idea of the evolution of free-piston engines at Sunpower.

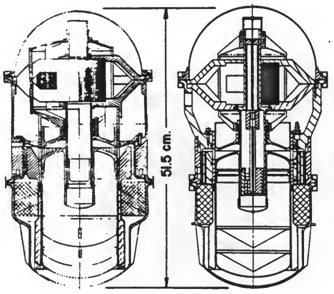

Figure 8 SPIKE 1 and SPIKE 2 to Same Scale

Table 1 Comparison of Free-Piston Engines Designed and Operated at Sunpower

	1973 Model 10	1974 AGA Type 1 - Type 3	1976 [illegible]	1976 RE-1000	1982 B-1	1982 B-2	1982 S-100	1982 SPIKE 1	1983 SPIKE 2
Power (Nominal)	10W	1-2kW	100W	1-3kW	2.5kW	10kW	100W	1kW	1kW
Measured Efficiency Based on PV Work and Heat Out	10%	25% Type 3	10%	32%	44% P- 38% D	44% P (NM)	25% P	36% P	33% P
Frequency (Hz)	14-20	25	35	30	30	50	50	60	60
Charge Pressure (bar)	6	70	16	70-80	50	20	5	8.5	8.5
Weight (kg)	10	105	15	160-350	25	150	15	23	30
Heat Source	elec.	elec.	charcoal(NR) gas(R)	elec. & gas (R)	gas(R)	solar	kerosene (R)	coal & gas (NR)	gas (R)
Hot End Heat Exchanger	Hex Cap	tubes	integral fins	tubes	tubes	flat collector	integral fins	integral fins	integral fins
Cooling	Water	Water	Water	Water	Water	Water	Air	Water	Water
Loading Device	IP & LA	IC	LA	DA & IC	DA & HP	LA	LA	LA	LA
Number Constructed	50	3	50	7	3	1	1	1	1
+Size (cm) Diameter Length	11 34	‡ 34.5 116	* 15 28	* 34.5 116	* 19.5 45	44 83	10.5 31.5	27 51.5	27 51.5

Key:

P - predicted
D - demonstrated at this time
NM - not measured
R - recuperative
NR - non recuperative

LA - linear alternator
DA - dashpot load
HP - heat pump (duplex arrangement)
IP - inertia pump
IC - inertia compressor

* Includes over-sized pressure vessel for housing the dashpot load

+ Excludes burner but includes loading device except where otherwise indicated.

‡ Dimensions of engine only - load device not included

Note: Chart does not indicate programs involving other applications of Stirling cycle technology, eg, domestic heat pumps, freezers, crank engines, free-cylinder engines and Ringbom engines.

Studies which did not involve fabricated hardware are also not indicated.

Development

The SPIKE 1 has been built and has undergone extensive testing. Though the testing program is at this stage incomplete, we have obtained better than 700 W (electrical) at 60 Hz. The engine was originally run on a coal burner as a demonstration of the Stirling engine's ability to use various fuels. Later the burner was modified to use natural gas since this was more convenient in the laboratory. The starting sequence is extremely simple: the head is heated until the hot end dome temperature reads 400°C, the alternator is then excited with a voltage of 30 V at 60 Hz. The engine will normally start after a few seconds of excitation.

There have been some development problems which were not anticipated at the planning stage. The most serious problem has been the difficulty in getting the piston to center adequately. It turns out that owing to the more expanded (or 'fatter') PV trace of a low pressure design, the centering force due to gas leakage at the center ports is often inadequate to maintain the piston in its desired mean position. This problem is presently being addressed by an experimental and analytical program. Other development problems have been related to the alignment and rigidity of the initial prototype. The alignment is a particularly important problem in that it impacts strongly on the wear of the rubbing surfaces. Many different bearto the cooler where the external surface area could facilitate air cooling directly on the engine, Air cooling has not been tried on our 1 kW engines, but a smaller 100W low pressure engine of similar proportion (the S-100) has run successfully with such a finned cooler. Figure 5 shows the 100W low pressure design, the geometry of the cooling fins is clearly visible. At this point both SPIKE 1 and SPIKE 2 employ water for cooling as this is considered more convenient during the development phase.

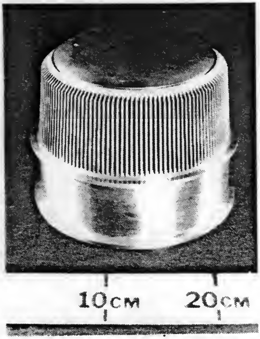

Figure 4 Head of SPIKE 2

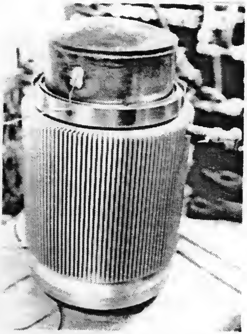

Figure 5 S-100 Showing Cooling Fins

The piston gas spring has been another area of concern. The loss associated with gas springs, referred to as gas spring hysteresis loss, may be quite significant. In the case of a relatively stiff piston gas spring this may be on the order of a few hundred watts. Typically the external gas spring supplies only part of the required spring rate, the rest being supplied by the working gas. It is therefore advantageous to reduce the requirement for an external gas spring as much as possible. For a given frequency, the required spring rate is directly proportional to the reciprocating mass. Thus, the lower the reciprocating mass, the smaller the spring rate may be. For sufficiently low reciprocating mass there is often no need for an external gas spring at all. Unfortunately, older generation linear alternators have required relatively heavy plungers. It was therefore expedient that we embark upon a parallel linear alternator development program. This program has produced a very compact moving magnet design with a minimum plunger mass of only 240 g. With the piston and connecting pieces the reciprocating mass for the new 1 kW designs is on the order of 1 kg. The mass of the stator is 7 kg and the electrical efficiency is better than 80% at 60 Hz and 3 cm stroke (Figure 6).

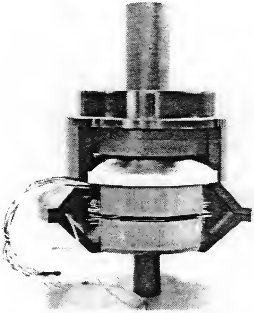

Figure 6 SPIKE Alternator

A potential problem that has had to be resolved is the tendency for low pressure designs to exhibit fairly high appendix gap losses (ie, the loss associated with the irreversibilities occuring in the gap between the displacer and its cylinder). Analysis has shown that by insulating the displacer cylinder from the regenerator and hot heat exchanger, it is possible to reduce this loss significantly. The purpose of the insulation is to allow the axial temperature gradient along the displacer and cylinder to relax. That is, the axial temperature gradient to become shallower. Theoretically, for zero temperature gradient the loss is zero, increasing for increasing gradient (6). In the present designs, our calculations show a re-

Since 1977 studies have been conducted to evaluate the RE-1000 technology for application to a preproduction prototype of similar power (4). In November 1980 work was started on a hardware program in which the design and development of a 1 kW electrical output alternator engine would be undertaken. Two generations of engines have been built under this program, the first one is a coal fired configuration called SPIKE 1 and the second machine is of similar design, embodying some modifications and improvements, called SPIKE 2. The SPIKE 2 is to be natural gas fired using a recuperative burner.

It should be noted that there are other hardware programs also directed at linear alternator technology which have evolved in a similar manner. Two such programs are the D-2 programs in which the target is a 10 kW solar engine and the S-100 program which is directed to a lightweight, hand portable 100 W engine using kerosene as fuel.

Design Philosophy

During the RE-1000 program it became evident that Stirling engine development involved a set of problems which would be critical in any effort at commercialization. There were at least three important areas that needed to be addressed, viz:

a) Tubular heat exchangers, while being perfectly adequate in our research engine, are expensive to fabricate. Obviously a cast head with integral fins would be an advantage. The high pressure engines (50 bar and above) are of small diameter thus making it difficult to configure integral external heat exchangers in the available space. Since the external heat transfer surface area would be relatively smaller, it would be necessary to use a combination of larger blowers and high surface temperatures. Both these requirments lead to an expensive construction often requiring super-alloy components. The advantage of the tubular heat exchanger is that its vastly superior surface area allows a much smaller blower and lower peak temperatures to be used. For example, a recently operated 2.5 kW, 50 bar engine with a tubular heat exchagner required only 10 W to the blower and the highest temperature on the fins was 700°C. The low power blower is very quiet which is a further advantage in many applications. However, the problem of expense associated with the complexity of a tubular heat exchanger is overriding in most applications.

b) The loss of power and efficiency and centering problems associated with leakage past sliding seals has been an ongoing difficulty with high pressure designs. Clearances required on the sliding seals have had to be quite small (often as low as 10 μm). With clearances of this size, the tolerance requirements on circularity, concentricity and squareness are much too severe and lead to expensive machining techniques. It was essential, therefore, to investigate ways in which these tolerance requirements might be relaxed.

c) Finally, it has been suggested that the high pressures used in Stirling engines may be seen as a potential hazard in a domestic environment. This point may be argued, but it could be said that consumer resistance to an 80 bar engine would be greater than to an engine of under 10 bar charge.

The Design

An early observation was that by designing for a much lower charge pressure, it is possible to reduce the cycle sensitivity to seal leakage. Figure 3 compares the seal leakage as an enthalpy loss from the compression space for two engine configurations - a high pressure design (RE-1000) and a low pressure design (SPIKE 1). It is evident that for a seal clearance of relatively large size, the low pressure design suffers only a small loss. Therefore, the tolerance on the clearance seal of a low pressure design is relatively easily met whereas that for the high pressure design could lead to manufacturing difficulties. An obvious suggestion is to use rings. However, rings tend to pump differentially and are quite unpredictable in the direction they may pump. Though this is acceptable to a degree in a crank machine, it does present a serious difficulty to a free-piston machine.

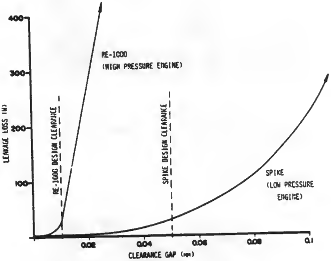

Figure 3 Sensitivity to Seal Leakage

Since power is proportional to the product of charge pressure, swept volume and frequency, a low pressure design will characteristically be larger in diameter and probably have a higher operating frequency than a high pressure design (5). This increased diameter of the engine may be advantageous to the incorporation of integrally finned external heat exchangers. Investigating this possibility, it was found that a 1 kW engine of 8.5 bar charge pressure had very favorable surface area geometry for integral finned heat exchangers. The heater surface area was such that a small blower of around 10 W could be used for combustion air circulation and, furthermore, the surface temperatures were sufficiently low for stainless steel to be used as the material for construction. Figure 4 shows the head design of SPIKE 2, note particularly that the configuration lends itself to investment casting quite readily. A similar argument applies ing configurations have been evaluated to develop a rational approach to subsequent designs.

The SPIKE 2 design, being a follow-on project, has been able to benefit strongly from the SPIKE 1 experimental program. For example, it was felt that effort should be made to reduce the size of the regenerator. SPIKE 1 has a large regenerator and this is potentially expensive in a production design. In fact, the low pressure designs tend to require regenerators of considerably larger size than that of the high pressure designs. A parametric study was conducted which concluded that the regenerator could be reduced in size without too great a compromise in efficiency. Other aspects of the SPIKE 2 design include a reduction in the number of stack-up components to alleviate the tolerance build up problem of the SPIKE 1, and an effort to improve the overall rigidity of the design. The SPIKE 2 thermodynamic design has been compromised somewhat (particularly efficiency) in the interests of hardware integrity. However, the expected overall efficiency should be better than 25% (electrical power to heat in). This performance is more than adequate for a successful commercial design at this power level.

Table 2 lists a comparison of the loss inventories and performance of the SPIKE 1 and SPIKE 2 engines, and Figure 9 shows the completed SPIKE 2 ready for testing.

Table 2 Loss Inventory for SPIKE 1 & SPIKE 2

	SPIKE 1	SPIKE 2
Piston Leakage Loss (W)	8.0	10.3
Displacer Spring Hysteresis (W)	130.0	126.8
Cooler Pumping Loss (W)	36.4	38.23
Regenerator Enthalpy Flux (W)	170.0	199
Regenerator Pumping Loss (W)	42	210
Heater Pumping Loss (W)	57.0	36.5
Displacer Conduction (W)	99.3	68.7
Displacer Cylinder Conduction (W)	100	70
Appendix Loss (W)	130	147
Wall Conduction (W)	142	102
Gas Conduction (W)	177	170
PV Power (KW)	1.4	1.4
Efficiency (%)	36	33

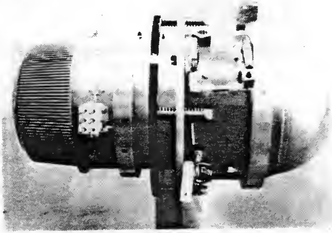

Figure 9 SPIKE 2 Ready for Testing

Conclusions

Sunpower has a long history associated with the development of free-piston Stirling engines. During this time it has become evident that low to medium pressure (8.5 to 25 bar) machines have many characteristics which are favorable to low cost manufacture. The SPIKE program has demonstrated that a compact, low pressure free-piston Stirling engine alternator set can be built and operated with acceptable performance and specific power. This has been a major departure from the high pressure design typically found in the industry. Two generations of the machine have been built, the first of which has demonstrated the feasibility of the approach. The second generation machine is aimed at refining the hardware and is now on test in our laboratory.

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Direct fired solid fuel Stirling engine

Efficient conversion of biomass to power

Declining oil reserves and uncertainty of foreign energy supply have produced an increasing interest in utilization of domestic fuels, e.g. biomass and coal.

Energy substitution corresponding to 5 % of the US oil consumption, could presently be supplied from forest and crop residues.

Local shortages and economic survival are motivating many industries to generate their own electricity, using less expensive and readily available energy.

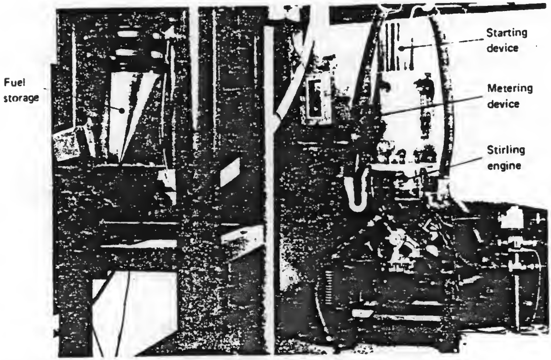
The Stirling engine – environmentally clean and silent – has the capability of effectively utilizing any liquid, gaseous or solid fuel.

United Stirling (USS) has on its program development of 40–150 kW engines for stationary equipment, e.g. generator sets.

Since conversion of solid fuels into gaseous or liquid form includes a considerable loss in thermal efficiency, USS has concentrated its solid fuel combustion program on direct fired systems.

Combustion rig tests have clearly demonstrated the ability to meter and burn different types and sizes of biomass, e.g. poplar powder (~100 mesh), saw dust (~6 mesh), chips (~2.5 mesh).

Recently a solid combustion system was adapted to a 40 kW Stirling engine. Initial tests with beech wood (6 mesh), have now been successfully performed.



Beech wood fired Stirling engine on test stand

KB United Stirling (Sweden) AB & Co
Box 856
S-201 80 MALMO
Sweden
Tel 040 10 09 50

United Stirling Inc
211 The Strand
ALEXANDRIA, VIRGINIA 22314
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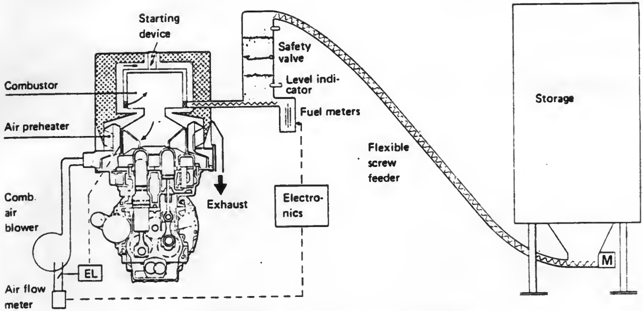
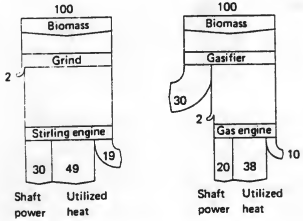


Basic principles for the solid fuel combustion control system

The upper size of biomass is presently limited to 2.5 mesh due to inaccurate metering of larger particles; but size will gradually be increased to reduce fuel pre-preparation work.

Gains in total efficiency from a direct fired Stirling engine, as compared to a conventional spark ignited engine running on gasified biomass, is shown in the figure below.



Energy flow charts for biomass fired Stirling and Otto engine

For field testing, modules of Stirling engine generator sets, ranging from 40–75 kW, could be tailored to meet varying industrial, agricultural or community needs.

Of specific interest are applications where inexpensive fuels are available and there is a need for electric power, e.g. lumber industries, forest and agricultural farms. In the long term, undeveloped countries with almost unlimited access to inexpensive fuels, represent a growing market.

Other markets include co-generation systems for residential heating and decentralized electricity production in cooperation with the local utility network, applied to cluster homes, hospitals, commercial buildings etc. The outstanding environmental qualities of the Stirling engine allow integration with existing buildings and systems. Energy savings with co-generation are in the order of 30 % as compared to conventional furnace heating and electricity generation at fossil fuel power plants.

To summarize, Stirling co-generation systems are

- Highly efficient, energy saving
- Clean – extremely low contents of CO, NO_x, HC, particulates
- Silent, vibration free
- Retrofittable
- Cost effective – competitive even at current energy prices.

789034

RESEARCH ON THE SODIUM HEAT ENGINE

T. K. Hunt, Neill Weber and Terry Cole

Research Staff - Ford Motor Company

ABSTRACT

The Sodium Heat Engine (SHE) is a new device for direct thermoelectric energy conversion. It uses the ionically conducting ceramic, β"-alumina, to form a high-temperature concentration cell for elemental sodium. The vapor pressure (activity) gradient across the cell is maintained by a high temperature heat source (at 600-1000°C) on one side of a β"-alumina membrane and a low temperature condenser (at 100-200°C) on the other side.

Theoretical analysis of the SHE shows that under quasi-reversible conditions the efficiency should be more than 90% of Carnot efficiency. For typical operating conditions of T_2=800°C, T_1=100°C, a specific power output of 0.7 watt/cm^2 has been achieved in test electrodes. If certain design criteria involving parasitic heat losses can be met, the SHE should achieve overall thermal efficiency in the range of 20-40% at a power output of 1 watt/cm^2.

THE SODIUM HEAT ENGINE (SHE) is a novel device capable of the direct conversion of heat into electrical energy. The SHE operates continuously as a concentration cell with the required concentration gradient maintained by the input heat source and output heat sink. In operation, sodium is expanded nearly isothermally through a beta-alumina solid electrolyte with electrical energy being extracted during the process. Theoretical analyses, combined with the measured parameters of operating sodium heat engines, predict an overall thermal-electrical conversion efficiency of 25-30% at 800°C for single cells. These calculations use component performance levels already achieved and do not assume additional bottoming cycles. The operating characteristics of the SHE, especially its high efficiency in small sizes, make it well suited for the local generation of electric power from chemical fuels or from concentrated solar energy, in a total energy system. Preliminary calculations of the overall efficiency of a central power station utilizing a SHE as a topping cycle show that for presently available design parameters a SHE conversion efficiency of 15-20% should be possible with the SHE operating between an upper temperature of 900°C and a lower temperature of 500°C.

The fundamental operating principles of the SHE have been described in detail in previous papers.(1,2)* A brief summary of these principles is presented here. Figure 1 shows a schematic diagram of the SHE. A closed container is partially filled with liquid sodium working fluid and is physically divided into high and low pressure regions by a pump and a tubular membrane of solid electrolyte which bears a porous metal electrode. The inner section of the device is maintained at a temperature T_2 by a heat source while the outer section is maintained at temperature T_1 ($T_2 > T_1$) by a heat sink. The temperature differential between the two regions gives rise to a sodium vapor pressure differential across the solid electrolyte membrane.

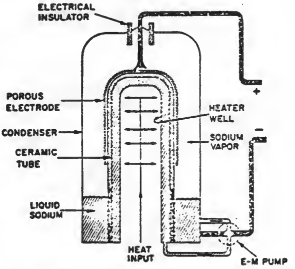


Fig. 1 - Schematic diagram of Sodium Heat Engine

During operation sodium travels a closed cycle through the device. Starting in the high temperature (high pressure) region, the heat source raises the incoming liquid sodium to temperature T_2. Since β"-alumina has high conductivity for sodium ions and negligible electronic conductivity, a mole of electrons must exit the high temperature zone for each mole of sodium ions entering the β"-alumina. Sodium ions then migrate through the β"-

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*Numbers in parentheses designate References at end of paper.

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alumina membrane in response to the pressure differential (gradient of Gibbs free energy). After passing through the external load, the electrons are recombined with sodium ions at the porous electrolyte interface. Neutral sodium then evaporates from the porous electrode at pressure P_1 and temperature T_2 passing through the vapor space to the condenser at temperature T_1. The condensed liquid sodium is then returned to the high temperature zone by an electromagnetic pump completing the cycle.

The processes occurring in the solid electrolyte and at its interfaces are very nearly equivalent to an isothermal expansion of sodium from pressure $P_V(T_2)$ to $P_V(T_1)$ at temperature T_2. Thus, the SHE is, in effect, a continuous concentration cell for sodium.

ANALYSIS OF SHE OPERATION

A thermodynamic analysis of an idealized device in which extraneous heat leaks from hot to cold reservoir have been ignored shows that the open circuit (zero-load) efficiency exceeds 90% of the Carnot efficiency. The efficiency of the SHE is the quotient of the net work output divided by the heat input. The output, under quasi-equilibrium conditions, consists of the work (W_1) done by isothermal expansion of the sodium from $P_V(T_2)$ to $P_V(T_1)$, less the work necessary to recirculate the liquid sodium W_2. The heat input per mole of sodium circulated includes the isothermal expansion work W_1, the heat of vaporization of sodium (L) and the enthalpy of sodium $\Delta H \simeq C_P(T_2 - T_1)$. Thus the efficiency

$$\zeta = \frac{W_1 - W_2}{W_1 + L + \Delta H} \qquad (1)$$

The following simplifying assumptions are made
(a)

$$W_1 >> W_2 \quad ; \qquad (2)$$

(b) Sodium behaves as a perfect gas, so the expansion work is given by

$$W_1 = RT_2 \ln \frac{P_V(T_2)}{P_V(T_1)} \qquad (3)$$

where $P_V(T_2)$ and $P_V(T_1)$ are the vapor pressures in the high and low pressure regions and R is the gas constant.
(c) The Clausius-Clapeyron equation describes the temperature dependence of Na vapor pressure

$$\ln \frac{P_V(T_2)}{P_V(T_1)} = \frac{L}{R} \left[\frac{T_2 - T_1}{T_2 T_1} \right] \qquad (4)$$

Using (2), (3) and (4) the efficiency may be expressed in terms of the Carnot efficiency $\zeta_c = (T2 - T_1)/T_2$,

$$\zeta = \frac{\zeta_c}{1 + \left[\frac{\zeta_c C_p T_1}{L}\right]} \qquad (5)$$

Eq. (5) gives a SHE efficiency of 46% for T_2 = 1000K and T_1 = 500K for which the Carnot efficiency would be 50%. Eq. (5) also shows that for this class of device, good efficiency is obtained by choosing a working fluid with a small value of C_P/L. For sodium $C_P/L = 3.4 \times 10^{-4} K^{-1}$.

The open circuit voltage is derived directly from Eq. 3 and is

$$V_{o.c.} = \frac{RT_2}{F} \ln \frac{P_V(T_2)}{P_V(T_1)} \qquad (6)$$

where F is the Faraday. This is the Nernst equation for a concentration cell, with pressure in place of activity or concentration..

A small correction to Eq. (6) is appropriate in the present case for which $P_V(T_1)$ is sufficiently small that atoms leaving the porous electrode and bound for the condenser and vice-versa make on the average no equilibrating collisions in the intervening space. For this case, which is of general application for the SHE, the Langmuir assumption (that the effective pressure at a surface is that pressure which would give a condensation rate from the saturated vapor equal to the evaporation rate at the surface) can be applied to both the condenser and the porous electrode which directly faces it. At open circuit the number of Na atoms leaving the porous electrode per unit area at T_2 is

$$\dot{n} = \frac{P_{eff}(T_2)}{(2\pi MR)^{1/2} T_2^{1/2}} \qquad (7)$$

and since there is no net current this must be equal to the number leaving the condenser,

$$\dot{n} = \frac{P_{eff}(T_1)}{(2\pi MR)^{1/2} T_1^{1/2}} .$$

The effective pressure $P_{eff}(T_2)$ at the porous electrode is then properly substituted for $P_V(T_1)$ in (6) above, yielding for the open circuit voltage

$$V_{o.c.} = \frac{RT_2}{F} \ln \left[\frac{P_V(T_2)}{\left(\frac{T_2}{T_1}\right)^{1/2} P_V(T_1)} \right] \qquad (8)$$

where the very small electronic conductivity of the solid electrolyte has been neglected. The correction Eq. 8 makes to Eq. 6 is small -- amounting to about 50 mV for typical operating conditions.

When current is drawn from the SHE, an internal voltage drop occurs due to the resistance of the electrolyte-porous electrode system. In addition, the Na concentration level at the porous electrode-electrolyte interface must increase above the level corresponding to the open circuit equilibrium in order that there be net evaporation from that interface sufficient to account for the current being drawn. Thus, under loaded conditions, the voltage is more accurately described by

$$V = \frac{RT_2}{F} \ln \left[\frac{P_V(T_2)}{\left(\frac{T_2}{T_1}\right)^{1/2} P_V(T_1) + \left(\frac{2\pi MR}{F^2}\right)^{1/2} T^{1/2} I}\right] - IR_o, \qquad (9)$$

where the second term in the denominator represents a current-generated pressure.(2) An additional small correction results from the fact that at high pressures there is an increased tendency for the Na vapor to dimerize and thus to depart from the "ideal gas" situation which leads to Eq. 9. This correction reduces the output voltage at a given current by about 0.8 to 1.7% in the range from 500°C to 1100°C. It should be noted that for low values of T_1 — below about 300°C and for values of the current density, $I > 0.1\ \mathrm{A/cm^2}$, the pressure term in the denominator of the logarithm is relatively small and the voltage-current relation is then given very closely by

$$V = A - \frac{RT_2}{F} \ln I - IR_o \qquad (10)$$

The quality of the agreement of Eq. 10 with experiment can be seen in Fig. 8 and will be discussed further in the experimental section.

The efficiency of the SHE when parasitic heat losses are considered is given by the relation

$$\zeta = \frac{W_1}{W_1 + L + \Delta H + Q_{loss}} \qquad (11)$$

If we are to make Eq. 11 explicit, we must itemize carefully and include the effects of the necessary electrical leads from hot device to cool load. These output leads must be optimized between two conflicting requirements in that they should present a low electrical resistance to the device to minimize I^2R losses in the leads and thus maximize the power delivered to the load but should present a high thermal resistance so as to minimize parasitic thermal conduction heat losses. Since the ratio of thermal to electrical conductivity of metals, as given by the Wiedemann-Franz law[3] depends primarily on temperature and not on the choice of material, it will be seen that an optimum choice for the lead electrical resistance can be made. A more complete expression for the efficiency of the SHE including the effect of lead resistance R_ℓ is given by

$$\zeta = \frac{I(V_o - ISR_\ell)}{I(V_o - \frac{ISR_\ell}{2} + \frac{L}{F}) + \frac{IC_p}{F}(T_2 - T_1) + \frac{\mathcal{L}}{2R_\ell S}(T_2{}^2 - T_1{}^2) + \frac{\sigma}{Z}(T_2{}^4 - T_1{}^4)} \qquad (12)$$

Here the ΔT^4 term in the denominator represents the radiation transfer from the hot porous electrode to the cool condenser of reflectivity $(\frac{Z-1}{Z})$ in the near infrared and σ is the Stefan-Boltzmann constant. The ΔT^2 term is the parasitic thermal conduction loss term expressed via the Wiedemann-Franz ratio $(K\sigma = \mathcal{L}T)$ in terms of the lead resistance R_ℓ and the Lorenz number $\mathcal{L} = 2.45 \times 10^{-8}$ watt/ohm deg^2. S is the active surface area of the device.

Figure 2 shows a family of power vs. efficiency curves at different temperatures T_2 with T_1 fixed at 200°C, with Z = 20, and for a fixed value of the specific internal resistance $R_o = 0.18\ \Omega\cdot\mathrm{cm}^2$.

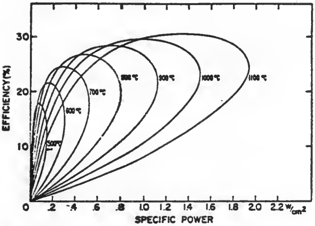

Fig. 2 - Calculated power-efficiency performance of SHE vs. T_2. $R_o = 0.18\ \Omega\cdot\mathrm{cm}^2$, $T_1 = 200°\mathrm{C}$, Z = 20.

In these curves the choice of R_ℓ in Eq. 12 has been optimized for maximum efficiency at each point on the curves. In comparison to fixed leads, this optimization procedure substantially extends the high efficiency operating region to both lower and higher powers. Figure 3 shows the variation of R_ℓ

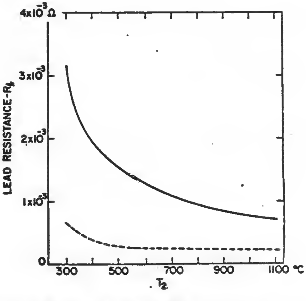

Fig. 3 - Optimum lead resistance: solid line - for maximum efficiency; dashed line - for best efficiency at maximum power point. $R_o = 0.18\ \Omega\cdot\mathrm{cm}^2$, $T_1 = 200°\mathrm{C}$, Z = 20, S = 100 cm^2

called for by this procedure both for the absolute maximum efficiency point and for leads optimized for efficiency at the maximum power point. It may be seen that maximum efficiency leads need only vary by a factor of 2 in the length to cross-sectional area ratio from 500°C to 1000°C, thus allowing for relatively simple design for a variable lead system, if desired. Figure 4 displays the maximum efficiency and the efficiency at maximum power as a function of temperature for the same device parameters used for Figure 2. At low T_2, it will be necessary to use substantially thinner electrolytes if the assumed $R_o = 0.18\ \Omega\cdot cm^2$ is to be maintained. β"-alumina membranes have now been produced at tube wall thicknesses of 100 microns.(4) These membranes would have R_o, exclusive of polarization, from $0.017\ \Omega\cdot cm^2$ at 800°C to $0.03\ \Omega\cdot cm^2$ at 300°C.

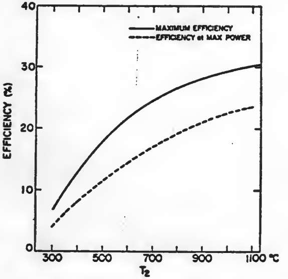

Fig. 4 - Efficiency vs. T_2, optimized leads. $R_o = 0.18\ \Omega\cdot cm^2$, $T_1 = 200°C$, $Z = 20$

To consider the SHE as a topping cycle, we calculate the effect of T_1 on the power-efficiency relation. Figure 5 shows a family of power-efficiency curves as a function of T_1 with T_2 fixed at 900°C and $R_o = 0.18\ \Omega\cdot cm^2$ as before. These curves were calculated for optimized leads. The monatonic increase in efficiency with decreasing T_1 is reversed between 200°C and 100°C due to the increased radiation loss and enthalpy terms in a temperature range where no useful improvement in $P_V(T_1)$ is possible.

Because it is, in principle, possible to series connect SHE cells at high temperature before bringing the output leads to room temperature, it is possible to drastically reduce the lead losses and improve the overall efficiency of the system to 30-35% at high power levels.

EXPERIMENTAL

At present, the major thrust of our efforts is directed toward understanding and improving the performance of the porous electrode system on β"-alumina. This work is most conveniently performed on demountable test cells (DTC) in which electrode systems can be readily interchanged and instrumentation easily introduced. In these cells provision for Na recirculation is not generally provided and a fully accounted overall efficiency cannot, therefore, be determined. Before discussing this work, we summarize the status of our work with fully recirculating test cells (RTC).

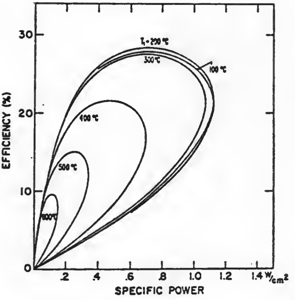

Fig. 5 - Calculated power-efficiency performance of SHE vs. T_1. $R_o = 0.18\ \Omega\cdot cm^2$, $T_2 = 900°C$, $Z = 20$

We have demonstrated efficiencies of 10% at 10 watts with porous electrodes capable of a maximum specific power of 0.16 watt/cm². Despite very poor wetting of the condenser surfaces by the condensing liquid sodium in those early experiments, values of $Z \simeq 10$ were achieved. A considerable variety of means are available to promote this wetting so as to provide a high reflectance specular surface and hence higher Z. We have shown that copper plating the condenser surface promotes good wetting but we have not yet had an opportunity to measure Z in such a cell. The electromagnetic pump for recirculating the Na is easily self-powered by passing the output current through it and this has been shown experimentally to require less than 0.3% of the output power. A completely sealed SHE system has operated for more than two weeks with no indication of need for a permanent vacuum system attachment. The porous molybdenum electrode is physically stable in the SHE environment at temperatures above 800°C for periods of over 1000 hours. We have also shown that these electrodes can collect current at 15 amperes per cm² without damage.

In order to more easily build and investigate the electrode properties of electrode-electrolyte systems, single-pass non-recirculating SHEs have been built. These are generally built in a geometry inverted with respect to our RTCs. A schematic of this arrangement is shown in Figure 6. A typical DTC header with an electrolyte-electrode system in place is shown in Figure 7. In this system, similar to the schematic of Figure 6, the β"-alumina tube is partially filled with sodium metal, a heater well-thermocouple well system inserted and sealed in place with an inert cover gas such as helium or argon. The cover gas reduces the heat leak to the upper end of the system, thus reducing the required heater power while protecting the elastomer seal at the open end of the β"-alumina tube from excessive temperatures. Without the cover gas, the evaporation and condensation of the sodium (heat pipe effect) would cause a very large heat leak. In the DTC system the pump is not included and because the supply of sodium per experimental run is limited, small electrodes, generally about 1 cm^2, are used so that high specific current densities may be drawn without depleting the Na supply too rapidly. As a practical matter, a SHE current of 1 ampere calls for a sodium throughput of about 1 cm^3/hour. With these small electrodes, it is also possible to produce and test a number of electrodes during each experimental run. The system shown in Figure 7 was used for three such electrodes. The copper shield surrounding the central β" tube was used to reduce radiative heat losses from regions of the tube not under test. We continue to find that thin molybdenum electrodes deposited on the β"-alumina by chemical vapor deposition (5) have excellent properties. Films formed by thermal decomposition of $Mo(CO)_6$ have high in-plane electrical conductivity, good Na vapor permeability normal to the plane, good adhesion to the electrolyte surface and good high temperature stability. These Moly films have a columnar structure with the grains perpendicular to the film plane, accounting for the useful porosity observed while the in-plane electrical conductivity appears to be within a factor of 2 of that expected for bulk molybdenum.

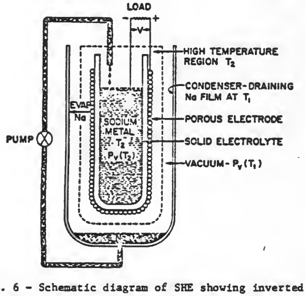

Fig. 6 - Schematic diagram of SHE showing inverted geometry used in demountable test system.

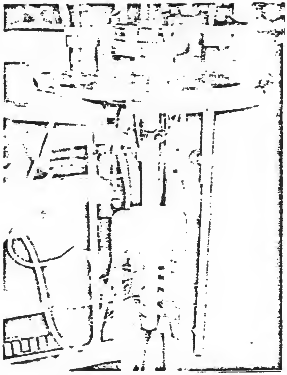

Fig. 7 - DTC with assembled SHE cell

The electrical contacts are made to the porous electrode using a harness of molybdenum wire wrapped with Cu wire and cinched in place with a small bolt. The Cu bonds to the molybdenum surface when the electrode reaches operating temperatures. For the purposes of the present work, copper wires are strapped under, and voltage probes spot-welded to the molybdenum harness. The DTC also contains a window through which the cell may be observed visually and the surface temperature monitored with an optical pyrometer.

In Figure 8 we show a family of experimental current-voltage curves for a single electrode as a function of temperature. This electrode was deposited on a 15-mm diameter β"-alumina tube supplied by the University of Utah.(6) The wall thickness under this electrode averaged 0.073 cm. The condenser temperature for this system is about 50°C since no recirculation is attempted. The solid lines through the data points are plots of Eq. 10 with the coefficients A and R_0 determined by a least squares fitting procedure. The fit is excellent over virtually the entire range as would be expected for a SHE having a very low T_1. This fitting procedure allows a rather precise determination of R_0 and of the A term in Eq. 10. The latter term is associated with the open circuit voltage parameters and with any resistance to neutral Na flow through the porous electrode. The

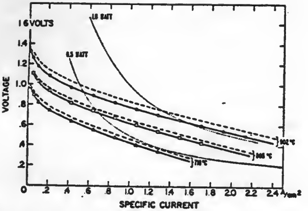

Fig. 8 - Experimental voltage - current curves as a function of temperature T_2

dashed lines are plots of Eq. 9 using as the only experimental input the value of R_o determined from the data fit discussed above. Thus, these curves display, with respect to the actual data, what is probably the net effect of Na flow impedance in the porous electrode. In this regard then, at 900° C the porous electrode in this test had a maximum power output within 10% of the ideal value for the experimental R_o. In Figure 9, we have plotted the values of A and R_o as a function of temperature for this same electrode. We also show the dependence of these coefficients on temperature for the case of an ideal electrode. The discrepancy in R_o between ideal and actual electrode performance is one measure of the "polarization" the major fraction of which is the interfacial impedance between electrolyte and porous electrode. It is this polarization which ultimately limits the improvement in specific power and efficiency to be gained by operating with thinner electrolyte membranes for which the bulk resistivity contribution is reduced.

We have also conducted experiments on a series of similar electrodes deposited on electrolytes of varying thickness produced by grinding a single ceramic tube to different diameters. The I-V curves produced in this way were analyzed for A and R_o and R_o then plotted as a function of the wall thickness δ. Such data are shown in Figure 10 for a temperature of 850°C. From the extrapolated zero thickness value of R_o, one may infer the polarization. The bulk resistivity of the electrolyte may be deduced from the slope of the line. The values of resistivity for the β"-alumina deduced in this way are in agreement to within 20% with those obtained by more direct means using a 4-terminal d-c approach.(7)

The power-efficiency temperature curves in Figures 2 and 5 were calculated using for R_o the value obtained, at 900°C, for the experimental cell whose I-V characteristics are shown in Figure 8. This value, $R_o = 0.18\ \Omega\cdot\text{cm}^2$ was achieved for a wall thickness of 0.073 cm. Tube walls of this thickness are easily fabricated of β"-alumina even by the batch process in principal use today. Green-

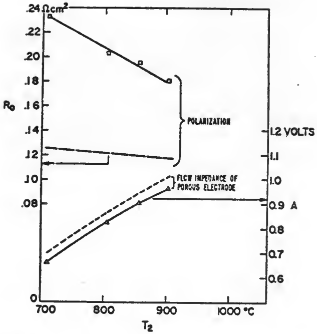

Fig. 9 - Coefficients of voltage - current relation for real (solid line) and ideal (dashed line) electrodes

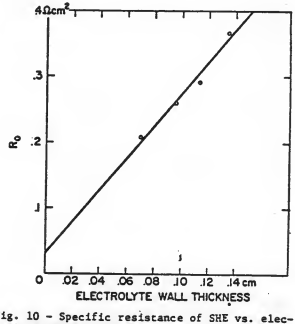

Fig. 10 - Specific resistance of SHE vs. electrolyte wall thickness. $T_2 = 850°\text{C}$

forming of β"-alumina tubes by extrusion has already shown the capability for producing dense high quality ceramic at wall thicknesses of 0.01 cm. With no further reduction in polarization beyond that presently achieved. Fig. 10 shows that a wall thickness of .01 cm would lead to $R_o \approx 0.05\ \Omega\cdot\text{cm}^2$

and specific powers and efficiencies substantially better than those displayed in Figures 2 and 5 above.

RECIRCULATING TEST CELL DESIGN

The small electrodes and batch filling scheme so useful for repeated experiments on the electrode-electrolyte subsystem are not suitable for the experimental determination of efficiency. For this, a complete recirculating system with electrode on all of the heated electrolyte area is required so that proper accounting of all the heat losses can be made and compared with the output power of the SHE. A photograph of a disassembled RTC with a nominal rating of 75 watts is shown in Figure 11. We plan to conduct tests of such cells in the near future.

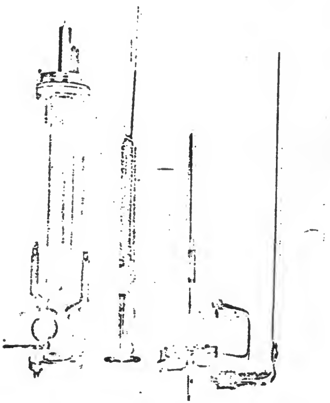

Fig. 11 - Experimental recirculating SHE - nominal 75 watt design

CONCLUSIONS

In our experiments, the Sodium Heat Engine has demonstrated the elements of operation needed to deliver electric power at a conversion efficiency near 25%, and with thinner electrolyte and series connection at the high temperature, the SHE overall efficiency should reach 35-40% with no further breakthroughs in electrode performance required.

Because the efficiency of the SHE does not depend on its size, it should be considered a prime candidate for distributed electric generation in residences, industrial plants and remote sites, especially in situations where the use of heat at the heat sink temperature is also called for. Little serious consideration has been given thus far to use of the SHE as a topping cycle with a steam turbine system for central station power generation, but the analysis of SHE operation suggests strongly that efficiencies of 15% can be achieved when the SHE exhaust temperature is as high as 500°C.

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7. N. Weber, T. Cole and T. K. Hunt, Manuscript in preparation.

THE SCROLL MACHINE—

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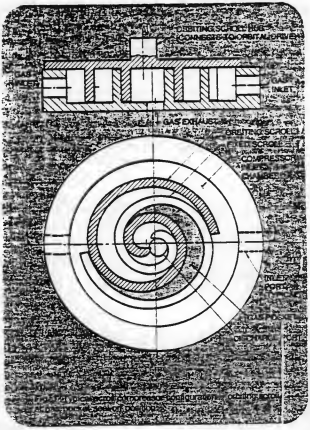
Arthur D. Little, Inc. (ADL) in Cambridge, Mass., is best known for its research and management consulting activities. Not much has been said, however, of an important ADL profit center that is concerned with developing and promoting inventions. Arthur D. Little Enterprises, whose predecessor, the Invention Management Group, was founded 20 years ago, generates an annual royalty income for ADL approaching $2 million—and has an enviable track record in commercializing its projects. The most recent product prospect—the scroll machine—promises to be another winner as it moves from the laboratory into the industrial mainstream via a series of manufacturing and marketing licensees.

The concept of a scroll machine—a design based on the interaction of two interleaved spirals or so-called "scrolls," with one scroll fixed and the other orbiting against it—probably traces its roots back to the liquid-moving ideas of Archimedes and Leonardo da Vinci. These primitive conceptions were reduced to practice in a scroll design that was first patented in the U.S. in the early 1900s. It then lay dormant again until about five years ago when the notion of a scroll compressor was brought to the attention of Arthur D. Little by an outside consultant-physicist, Neils Young, who apparently saw the practical possibilities for a scroll machine.

Arthur D. Little Enterprises receives some 4500 unsolicited invention disclosures a year, perhaps 10 percent from in-house and 90 percent from the outside. After screening and evaluation, approximately 30 (maybe ⅓ from Arthur D. Little, ⅔ from the outside) reach the project stage. While Arthur D. Little as a matter of policy does not pay its inventor-clients any front money, it does invest its own funds in taking an inventor's promising brainchild a step or two further in preparing it for licensing. Arthur D. Little's contracts with inventor-clients normally provide for a 50/50 split of licensing revenue. An initial expenditure by Arthur D. Little of $10–50,000 in developing a promising product or process is typical. The scroll machine, which was evaluated and developed for several applications to date, has cost Arthur D. Little in excess of $300,000. This sum covers patent protection, the design, building,



[illegible] typical scroll compressor configuration—orbiting scroll [illegible] seal-off position [illegible] and testing of prototypes, market research, and Arthur D. Little's marketing and negotiating efforts with carefully selected manufacturers.

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[1] Manager, Scroll Program, Arthur D. Little, Inc. Mem. ASME.
[2] Contributing Editor, ME. Mem. ASME.

The development effort at Arthur D. Little has resulted in commercially attractive scroll compressors for helium, air, and refrigerants. In addition, Arthur D. Little has also successfully completed work on a competitive scroll liquid pump for automotive fuel pumping applications. As a result of these programs—as well as Arthur D. Little-sponsored laboratory investigations—there is convincing evidence at hand to substantiate the belief that scroll machines offer commercial potential for a variety of gas handling and liquid pumping requirements. Positive displacement, self-priming, and good efficiency are among the desirable operational features of this type of liquid pump. Scroll pumps can be made from any suitable material that is compatible with the working fluid including plastics, graphites, or ceramics. By using appropriate materials and design characteristics, this type of pump can be operated at high temperatures and pressures.

The basic elements in a scroll machine are two identical spiral scrolls, each fixed to (or an integral part of) a cover plate. When the two scroll plate assemblies are mated, the two scrolls are interleaved in an eccentric fashion so that they touch and form a series of crescent-shaped pockets. The two cover plates seal these pockets, thus serving roughly the same function as cylinder walls in a reciprocating compressor.

One of the scroll plate assemblies is fixed. The other one orbits around the center point of the fixed scroll plate assembly, and, as it moves, the pockets formed by the interleaved scrolls follow the spiral toward the center and diminish in size. (If the device is used as an expander, the process is the reverse, i.e., the pockets move from the center to the periphery and enlarge.) The moving scroll plate assembly orbits with a fixed angular orientation; it does not rotate. It is driven by a simple short-throw crank assembly incorporating a radial compliance drive feature. Figure 1 shows the typical scroll plate assembly configuration; and Fig. 2 shows the progress of the compression process.

Operation

When used as a compressor, the inlet for the scrolls is at the periphery. The entering gas is trapped in a

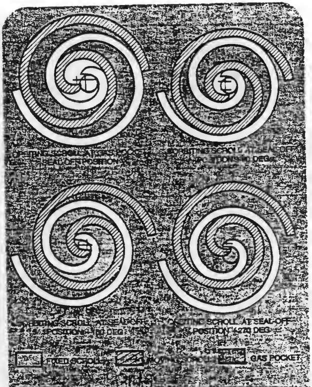
ORBITING SCROLL AT GAS POCKET SEAL-OFF POSITION
ORBITING SCROLL AT SEAL-OFF POSITION + 90 DEG
ORBITING SCROLL AT SEAL-OFF POSITION + 180 DEG
ORBITING SCROLL AT SEAL-OFF POSITION + 270 DEG
FIXED SCROLL
MOVING SCROLL
GAS POCKET

Fig. 2 Four operating positions of [illegible] scroll compressor/expander.

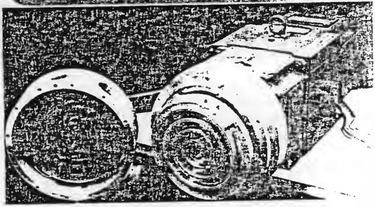

Fig. 3 A photo of laboratory prototype 3-hp ADL scroll compressor/expander with the compressor portion disassembled. The fixed scroll element has been detached and is shown to the left.

4 rotations of scroll

TABLE 1 A Summary of Performance Test Results on Laboratory Prototype 3-hp ADL Scroll Compressor/Expander

Working Fluid	Inlet Press (psig)	Inlet Temp. (°F)	Discharge Press (psig)	Discharge Temp. (°F)	Flow Rate (scfm)	Shaft Power (hp)	Average Efficiency, as a Percentage			
							Volumetric	Mechanical	Isentropic	Overall
Air	0.0	72	90.0	286	5.28	0.91	98	92	104	96
Helium	1.6	75	50.0	167	6.61	0.78	99	81	138	112

Notes

All tests were conducted on demonstration scroll compressor shown in Fig. 3 running at nominal speed of 1800 rpm.
Water cooling was used during air and helium tests.
Overall efficiency based upon isentropic compression; see *Compressed Air and Gas Handbook*, 3rd Ed., for definition of efficiency terms.

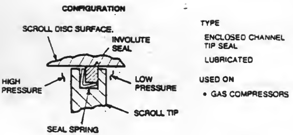

Fig. 4a Scroll seal techniques.

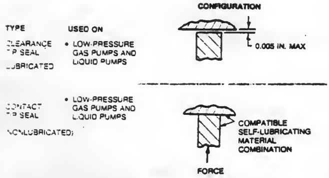
CONFIGURATION
TYPE
USED ON
CLEARANCE TIP SEAL
LUBRICATED
• LOW-PRESSURE GAS PUMPS AND LIQUID PUMPS
0.005 IN. MAX
CONTACT TIP SEAL
(NONLUBRICATED)
• LOW-PRESSURE GAS PUMPS AND LIQUID PUMPS
COMPATIBLE SELF-LUBRICATING MATERIAL COMBINATION
FORCE

Fig. 4b Scroll seal techniques continued.

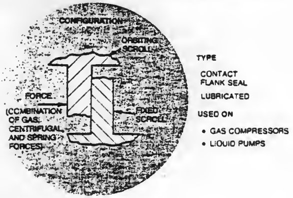

Fig. 4c Scroll seal techniques continued

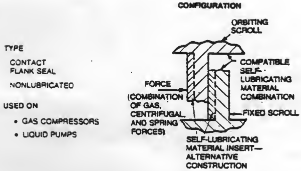
CONFIGURATION
ORBITING SCROLL
TYPE
CONTACT FLANK SEAL
NONLUBRICATED
USED ON
• GAS COMPRESSORS
• LIQUID PUMPS
FORCE
(COMBINATION OF GAS, CENTRIFUGAL AND SPRING FORCES)
COMPATIBLE SELF-LUBRICATING MATERIAL COMBINATION
FIXED SCROLL
SELF-LUBRICATING MATERIAL INSERT—ALTERNATIVE CONSTRUCTION

Fig. 4d Scroll seal techniques continued.

pocket and compressed as the pocket moves toward the center and diminishes in size. The compressed gas is exhausted through the outlet at the center of the fixed spiral. When the device operates as an expander, or Brayton cycle engine, the inlet becomes the outlet, the outlet becomes the inlet, and the moving scroll plate orbits in the opposite direction.

The displacement of a scroll machine is a function of the volume of the pockets formed between the scroll pieces. The volume of the pockets, in turn, is related to the parameters of the scrolls, such as the outside diameter, pitch, height (i.e., axial length) and thickness. The pressure ratio is directly related to the initial and final volume of the sealed pockets. For a scroll compressor, where the gas flows radially inward along the scroll, the final volume is also influenced by the discharge port configuration. Precise functional relationships between the displacement, the pressure ratio, and the scroll parameters have been completely defined. ADL has developed and is using on a regular basis computer programs that size and draw elements at any angular orbit position.

Scroll Performance

A laboratory prototype 3-hp (2.25-kW) demonstration scroll compressor was constructed early in the development program and tested to show typical performance capabilities. Figure 3 illustrates this unit with the compressor portion disassembled. The fixed scroll element has been detached and is shown standing at the left of the photograph. The orbiting scroll is connected to the drive motor. The drive motor is a special variable-speed, 1- to 3-hp (0.75–2.25-kW) unit selected for convenience in laboratory testing and is not representative in size of production-type compressor drive motors. The scroll compressor assembly is bulkier than the final production machinery would be since it was designed as an engineering test bed device. The scroll specifications are:

Scroll pitch	0.818 in.
Scroll axial length	0.650 in.
Scroll wall thickness	0.163 in.
Scroll outside diameter	6.100 in.
Orbit radius of moving scroll	0.246 in.
Volume ratio	3.70
Displacement	5.116 cu in./rev

The results of performance tests on this unit using various working fluids are summarized in Table 1.

Design Considerations

Typical Arthur D. Little, Inc. scroll machines incorporate standard hardware components like crankshafts, bearings, seals, and housings together with scroll components such as a fixed scroll, orbiting scroll, and a scroll coupling element. The scroll components require special design procedures which Arthur D. Little, Inc. has developed over the past several years. These procedures cover the mathematical definition of the scroll surface, the linear length of an involute spiral, the volume of the pocket formed between the interleaved scrolls as a function of its location, the center of gravity

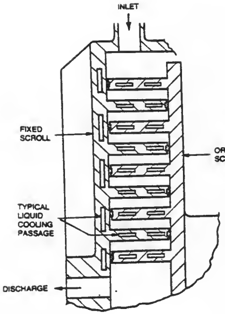

Fig. 5a Liquid cooling passage design.

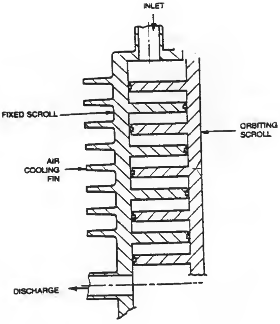

Fig. 5b Air cooling fin design.

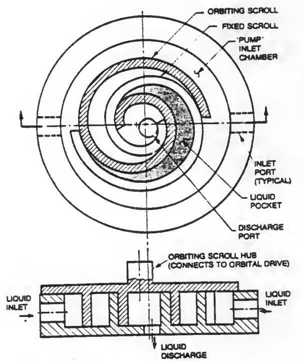

Fig. 6 A typical scroll liquid pump configuration — orbiting scroll at pocket seal-off position.

of an involute wall, and discharge port design techniques. In addition, procedures have been developed to compute the forces acting on a scroll component, such as the gas loading forces, the friction forces, and the centrifugal force.

Computer programs are frequently used to generate pictures of interleaved scrolls at any desired orbital position. Starting with typical parameters such as pitch—wall thickness angles for involute beginning and end—complete pictures of the cross section of the interleaved scrolls are drawn at desired angular orbit and scale (full-size, enlargement, or reduced size). Computer programs are utilized in similar fashion to determine the effect of leakage within the scrolls (given specified gap sizes at the tips and flanks of the scrolls) on overall machine performance.

All of this means that, of the many and varied scroll machines that ADL has developed to date, the design of the scroll components has become a well-defined and proven art. For instance, sealing techniques appropriate to scroll units have been evolved for lubricated and nonlubricated machines for handling gases as well as liquids. Several general examples of these techniques are shown in Figs. 4a through 4d. In addition, variations of these basic tip seal configurations have been adapted to minimize manufacturing costs. Designs have also been conceived to accommodate the cooling requirements of scroll gas compressors. Two schemes for air and liquid cooling of scrolls are shown in Figs. 5a and 5b.

Fabrication and Inspection

The components of a scroll machine that require special fabrication methods are the fixed and the or-

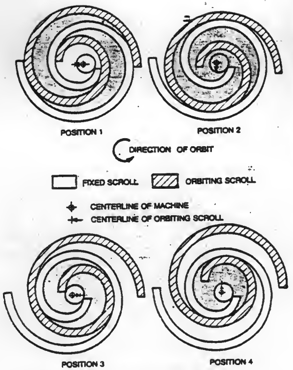

Fig. 7 Scroll pump operating cycle.

TABLE 2 A Summary of Scroll Manufacturing Techniques

Prototype Quantities

- Milling of scroll involute and disk surfaces from rough cast or bulk material using geared involute generation or numerical control technique

Production Quantities

- Gang milling of multiple scrolls on numerically controlled or cam controlled machine starting with precision cast parts
- Injection molding (for plastic parts) followed by light finishing of scroll tip surface
- New processes, to be developed, based upon stamping, coining, or equivalent techniques.

Required Tolerances

	Less than 6-in. scroll dia.	Over 6-in. scroll dia.
• Involute contour	±0.001 in.	±0.002 in.
• Disk surface flat within	0.002	0.005

Required Finish

• Involute surface	64 micro-inch RMS
• Disk surface	32 micro-inch RMS

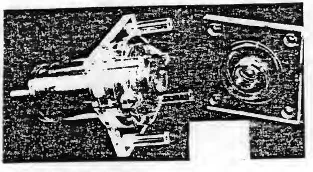

Fig. 8 A demonstration scroll liquid pump.

biting scroll parts. Since the scroll contour is involute in form, it is easily and perfectly generated by coordinated rotation and traversing of the part relative to a cutter. The necessary coordination can be provided by gearing or by N/C techniques. A listing of special machining processes, tolerances, and finishes for the scroll parts is given in Table 2.

Inspection of parts containing involute wall features can be readily accomplished using a surface plate and a vernier height gauge. The scroll wall configuration is specified by its intersection with orthogonal axes which meet at the part centerline. By measuring the distance between the scroll part centerline and the scroll flank surface parallel to these axes, the scroll accuracy and placement on the part can be fully determined.

Scroll Liquid Pump

A scroll liquid pump utilizes two involute shaped walls of 1.5 wraps each, which are integral with mounting disks so that they comprise right- and left-hand parts as shown in Fig. 6. As one scroll orbits—or moves around within the other without rotating—a continuous seal is maintained between the central region and the periphery, but no compression occurs. The fluid inlet port can be located at the center or at the periphery, depending upon the orbit direction and the design of the pump. For submerged pumps, the central port is usually the inlet with discharge at the periphery. Other applications where the pumped fluid is confined to the scroll area would use a peripheral inlet and a central discharge. This latter configuration is illustrated in Fig. 7.

Figure 8 shows a demonstration scroll liquid pump. The orbiting scroll is driven by a simple, adjustable throw-crank. Its angular alignment is maintained by the circular Oldham-coupling ring located directly behind it. Sealing is accomplished by controlling axial and radial clearances between the working surfaces of the two scrolls.

Scroll Gas Transfer Pump

Gas can be pumped with a hermetically sealed scroll pump which is driven by a magnetic coupling. All pump bearings consist of self-lubricating materials which are compatible with the process gas. The scrolls, which are machined from carbon-graphite, are lightly spring-loaded in the axial and radial direction for sealing. During operation, as the gas pressure across the pump increases, it exerts additional force on the scrolls

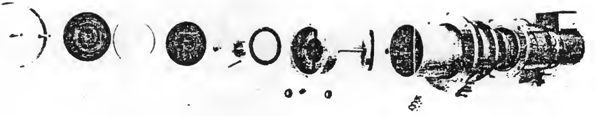

Fig. 9 A photograph of a laboratory model of the scroll gas transfer pump.

to enhance sealing. In this way, the forces on the scrolls that promote sealing are proportional to the overall pressure differences across the pump.

The scrolls are orbited by a crankshaft equipped with a pivoting spring-loaded bearing stud so that the orbit radius is free to accommodate to the optimum scroll operating orbit radius. A scroll coupling made from polyimide maintains the angular alignment of the fixed and orbiting scrolls. Finally, a circular metal thrust ring located between the scrolls at their periphery carries the compressive loading exerted on the scrolls by the differential gas pressure that is developed. Figure 9 shows a laboratory model of the scroll gas transfer pump.

Advantages

Because the compression process involved in the scroll compressor/expander can be designed to approach isothermal, overall compressor efficiencies of 90–95 percent can be attained. And because there are no valves, volumetric efficiencies of 100 percent are readily achieved. Flow rate through the compressor is little affected by discharge pressure, and flow volume per revolution is constant over a wide range of operating speeds.

Scroll compressors offer many mechanical advantages; chief among them is the absence of valves, which does away with the major item in compressor maintenance. Since these compressors have lower starting torques and more uniform driving torques than do reciprocating compressors, they can be driven by smaller motors. Scroll devices can be constructed of self-lubricating materials to provide oil-free operation. Operating experience confirms that scroll plates tend to "wear in" rather than wear out.

The design is simple, with few moving parts. Scroll devices should be less expensive to manufacture than reciprocating devices, and the design can be packaged in more compact configurations. Since they are continuous flow and completely balanced devices, essentially noise-free operation is possible. Performance test results on Arthur D. Little's laboratory prototype, a 3-hp (2.25-kW) compressor, confirm the practicality and inherent advantages of the design. Arthur D. Little is currently developing commercially attractive scroll gas compressors and scroll liquid pumps. Additional work is under way to adapt the concept to cryogenic applications.

A key advantage of the scroll gas transfer pump is long life. The relative rubbing speed between the scrolls is quite low (generally under 600 ft/min, or 180 m/min) since the pump operates at normal motor speeds but at a very small orbit radius. The maximum pressure loading on the scrolls is also very low. These factors lead to minimal wear of the scroll materials and therefore long life. In this machine, any wear which occurs tends to improve sealing of the scrolls because of the basic configuration of the parts and the axial and radial compliance of the driven mechanism.

Another advantage is reliable operation. There are few moving parts in the gas transfer scroll pump. No valves are needed in this pump either, since the action of the scrolls performs the valve function as well as the prime function of compressing the gas. All of these factors promote reliable operation.

The scroll gas transfer pump is a positive displacement device with high volumetric and compression efficiency. Furthermore, the continuous flow characteristics of the unit (it is not cyclical in operation) give it a versatile efficient operation that makes it especially useful for applications in the subatmospheric pressure region.

Commercialization

When Arthur D. Little Enterprises has brought an invention to the stage where its feasibility can be demonstrated, it looks for industrial licensees to complete the development, and to manufacture and market it under the terms of a royalty-bearing license agreement. So far, 13 seprate and distinct areas of exploitation for the scroll machine have been identified: air compression; refrigerant compression—stationary; refrigerant compression—vehicle: vacuum pumping; liquid pumping—vehicle; liquid pumping—stationary, medical; liquid pumping—marine; liquid pumping—general and chemical processing; special gas handling—helium compression; special gas handling—hydrogen isotope pumping/transfer; special gas handling—uranium hexafluoride pumping; hot gas expanders—Rankine cycle expanders; hot gas expanders—Brayton cycle expanders.

Nine licensees (who cannot be identified at this time for confidentiality reasons) have already been signed up by Arthur D. Little, and other agreements are being negotiated. Walter J. Cairns, vice president of Arthur D. Little and manager of Arthur D. Little Enterprises, says: "We expect great things from the many possible applications of the scroll machine in the years ahead, and we are particularly proud that it was Arthur D. Little that helped take an old principle and breathe new life into it!"

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